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ECL 61

STANFORD LINEAR ACCELERATOR CENTER II (A)

Beam Pipe Cooling System

Al Lisin, a mechanical engineer with an Engineer's degree from Stanford University and five year's experience with Aerojet-General Corporation in San Ramon joined the Mechanical Component Design group at the Stanford Linear Accelerator Center (SLAC) near Palo Alto, California, in 1961. He began work on the cooling of the accelerator beam pipe. There was already general agreement that the pipe would have to be cooled by a circulating liquid, but in other respects the form of the cooling system remained to be determined.

Prepared in the Design Division of the Mechanical Engineering Department, Stanford University, by John A. Alic with financial support from the National Science Foundation.

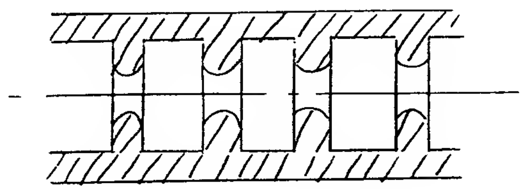
Background

The SLAC project had been proposed in 1957. For a number of years engineers and scientists had been carrying out studies and performing experimental work with much smaller linear accelerators at Stanford under government contracts. Detail design of components for the two mile linear accelerator started in 1961, and actual construction began in July 1962. Estimated total cost for the facility, design and construction of which was carried out by Stanford for the A.E.C. is $114 million. The accelerator began preliminary Stage I operation in 1966. Stage II operation, in which the electrons would be accelerated to greater energies, was planned for several years later. Some of the specifications of the accelerator are listed below:

	Stage I	Stage II
Accelerator Length	9,600 ft	9,600 ft
Length Between RF Power Feeds	10 ft	10 ft
Number of Beam Pipe Sections	960	960
Length of Sections	10 ft	10 ft
Number of Klystrons	240	960
Peak Power per Klystron	6-24 Mw	6-24 Mw
Beam Pulse Repetition Rate	1-360 pps	1-360 pps
RF Pulse Length	2.5 μsec	2.5 μsec
Electron Energy, Unloaded	11.1-22.2 Bev	22.2-44.4 Bev
Electron Energy, Loaded	10-20 Bev	20-40 Bev
Peak Beam Current	25-50 ma	50-100 ma
Average Beam Current	15-30 μa	30-60 μa
Average Beam Power	0.15-0.6 Mw	0.6-2.4 Mw
Accelerator Vacuum	10^{-5} mm Hg	10^{-5} mm Hg

Electrons are accelerated down the beam pipe by RF energy supplied to each pipe section by the klystrons. Every 10 foot section of pipe has a waveguide at each end for entrance or exit of the RF energy. The klystrons are a form of traveling wave tube. For Stage I each klystron feeds four beam pipe sections while for Stage II there will be one klystron per section. The 2 mile pipe comprising the accelerator itself is contained in a tunnel 25 feet below the klystron gallery, which is at ground level and contains most of the equipment requiring maintenance, as shown in Exhibit 1. An aluminum tube 2 feet in diameter supports the beam pipe.

The accelerator sections are made of copper with an O.D. of 4 inches and a wall thickness of 3/8 inch; they contain cavities inside of which the RF pulses resonate. The pipe is sketched below in longitudinal section:



There are 86 cavities in each 10 foot section. A considerable amount of RD power is dissipated in the pipe; this takes the form of heat generated on its inside surface. When Al Lisin began working on the cooling of the pipe, it had not yet been decided whether it would be of the "constant structure" type or the "constant gradient" type. In a constant structure accelerator (also termed constant attenuation), each cavity in the pipe is identical in size and the heat generated in each section decreases exponentially from the inlet end. A constant gradient structure has uniform heat generation along each section while the size of the cavities decreases from the inlet end to the exit end. The decision on which structure to use would be based on considerations of electronics and physics. Total heat generation in a 10 foot section for Stage II operation with either structure is a maximum of 13.4 or 13.6 kw, depending on whether the waveguide losses are included. At Stage I only 3.2 kw must be removed. Since Al did not know which form the accelerator would take, he had to consider cooling systems applicable to each at the Stage II condition.

The beam pipe cooling system is a critical part of the design of the accelerator because the cavities in the pipe must be dimensionally stable within ± .00005 inch. Al said, "I was told that they wanted to hold the size of the cavities constant within this tolerance. The actual size of the cavities would be set in tuning operations after fabrication. Tuning the accelerator is done by changing the sizes of the cavities slightly so the cavity will resonate. They hadn't then decided how the tuning would be done, although one method we were considering was to squeeze the cavity inward from the sides. If the dimensions of the cavities change once they have been set, there will be a shift in the phase of the RD energy pulses, putting them out of phase with the beam pulses. When making the sections of pipe we knew the tolerances on the cavities before tuning would have to be held to a few tenths of a mil. Actually, we'd hoped at first to be able to manufacture the pipe with sufficient accuracy for good RF characteristics, but then experiments showed the need for tolerances of 50 millionths and we knew the tuning operation would be necessary. This also meant we had to try to hold the temperature of the pipe constant to ± .2°F."

Since the amount of heat to be dissipated could be any amount up to 13.4 kw per 10 foot section, a control system to take account of this variation also had to be planned.

Beam Pipe Fabrication Studies

During the early phases of the accelerator project two methods of fabricating either a constant gradient or a constant structure pipe were considered -- brazing and electroforming. A pipe could be made from cup-shaped pieces of copper, separated by thin washers of brazing alloy and assembled into a 10 foot stack, then brazed. The electroforming process involved plating 3/8 inch of copper onto an assembly of alternating copper discs and aluminum spacer rings placed on a stainless steel mandrel. The mandrel would be removed and the aluminum etched out, leaving cavities. With either method the pipe would have to be made from oxygen free, high conductivity (OFHC) copper, with finishes on the internal surfaces of a few microinches to minimize RF losses. The OFHC copper is used because high conductivity is important for efficiency and because with the high purity material there is less chance of contaminants which might embrittle the structure. Also, if the copper were not oxygen free, oxygen and hydrogen might combine inside the pipe wall to form water vapor. This could result in blistering or excessive dimensional changes in the copper.

The progress of the fabrication studies can be traced through excerpts from SLAC Status Reports, published quarterly. From the 1 January 1960-30 June 1960 report:*

B. Brazing

An empirical evaluation of brazing cavity-sized parts together to form a ten-foot section of disk-loaded waveguide is being carried out. The parts for this assembly have been machined from blank cups which were made by stamping, drawing and piercing OFHC copper plate 3/8" thick by 6" wide.

A total of 125 of these cups have been drawn; 25 of these have been machined to finish tolerances. These finished cups are now being used to make short sections for evaluation of various brazing methods. Figure 2 displays a drawing of the blank cup after drawing, and Figure 3 displays a cup that has been machined to finished dimensions.**

* LINEAR ELECTRON ACCELERATOR STUDIES, M.L. Report No. 741, August 1960, Stanford University.

** Figures 2 and 3 appear in Exhibits 2 and 3.

The brazing experiments completed to date have been carried out using two cup assemblies arranged to simulate the conditions to be encountered when ten-foot lengths are brazed.

The results from these brazing experiments have been very satisfactory. The dimensional changes that have occurred during brazing have been insignificant, and satisfactory fillets have been obtained inside the cups. The axial dimension change has been less than .0001" during brazing.

Exceptionally good surfaces have been obtained on all internal surfaces of the cup by a lapping technique. The surface finish is 4 to 6 microinches RMS. The lapping experiments have just been started and it is too early to evaluate either the cost or the rf characteristics of the surfaces.

In addition to the experiments with the drawn cup shown in Figure 2, we are evaluating techniques of making the blanks by forging. To date we have evaluated a dozen such forgings, and the results of our tests indicate that the forging technique yields perfectly acceptable cups if it is properly done.

C. Electroforming

In order to carry out various electroforming experiments that can be fully evaluated from a microwave standpoint, additional parts have been started in process. A number of forged aluminum spacers have been ordered in order to evaluate the forging approach of making spacer blanks. The per-unit cost of the forging in a quantity of 1,000 is less than the cost of the solid bar stock required to machine the aluminum spacers.

A lapping technique that yields surface finishes of two to four microinches is being evaluated. A half dozen spacers have been lapped and are now being processed. At this point it is too early to tell whether or not the lapped part can be successfully processed through the various plating operations. At the time of writing this report it does, however, look very promising.

A new small plating facility is being established to evaluate ideas which appear to have the potential of reducing the cost of electroforming substantially, and at the same time yield higher quality copper. The facility is expected to be completed within the next 60 days. Among the problems to be studied with our new small plating facility will be those pertaining to solution concentration, agitation, rates of plating and related factors such as filtration rate.

At the present time we are electroforming samples on a stainless steel mandrel in a 30-gallon bath prepared with reagent grade copper sulfate, CP sulfuric acid and distilled water. Samples will be electroformed using current densities of 5, 10 and 15 amps per square foot. The samples will be given the usual mechanical and chemical tests.

A 30-gallon copper fluoborate bath was set up and samples plated. It was found that very high current densities could be used and a fairly smooth deposit obtained but the deposit had a long columnar grain structure that yielded inferior mechanical characteristics.

Spectrographic analysis of the Stanford water and accumulated sludge on a five-micron filter indicates that the water is very high in iron content and is therefore not suitable for plating. Consequently, a still has been installed to provide distilled water at a rate of 20 gallongs per hour for plating applications in the existing plating shop.

From the 1 July to 30 September 1960 report*:

A. Brazing

Several more brazing test have been made with the cavity-sized cups. These tests have yielded the same good results that were reported in the last Status Report (M.L. 741).

The cups made by fórging techniques worked out so well that 300 forged cups have been ordered and received. These cups are now scheduled for machining.

No additional work has been done with the lapping of the internal surfaces of the cups. It was previously determined that excellent surface finishes on the cups could be obtained using a lapping technique, but it appears at the present time to be uneconomical to lap the cup geometry unless new methods can be found.

B. Electroforming

The new small plating facility reported in the last Status Report is now expected to be in operation around November 1.

The spacers that were lapped to the two-to-four microinch surface finish have been processed and found to be completely compatible with the process procedures. Five of these lapped spacers were clamped together to form a simple aluminum cylinder. The section was then processed in the same manner as accelerator waveguide. Profilometer readings on the cylinder I.D. gave a surface finish of two to four microinches, indicating that the lapped surface smoothness was retained through electroforming, subsequent etching and chemical plating. The superior surface finished obtained from the lappint techniques appeared to justify further consideration particularly since the cost of lapping both the disk and spacer is less than 10% of the total estimated cost of these two parts.

The 1,000 aluminum forgings have been received. A number of these forgings have been finish-machined and processed, with good results.

A forged spacer is shown in Exhibit 4.

* M.L. Report No. 770, M Report No. 232, November 1960, Stanford University

By January of 1962 a furnace suitable for brazing a 10 foot accelerator pipe section held vertically had been completed. An electroplating unit suitable for 10 foot sections had also been installed. The 1 January to 31 March 1962 Status Report* then stated, "Three 10-ft dummy sections have been electroformed, and two constant-gradient disk-loaded waveguide sections have been brazed. The sections were found to be vacuum tight and are now undergoing rf tests."

From the 1 April to 30 June 1962 report**:

B. Electroforming and Brazing

Five 10-ft sections were made during the period (two electroformed and three brazed). These initially have been used in the evaluation of electroforming and brazing techniques. After careful consideration we have come to the conclusion that we should fabricate the disk-loaded waveguide by the brazing method. While we have every reason to believe that no difficulties will arise, the importance of this component to the project warrants care in planning. Therefore, we intend to retain the electroforming capability on a standby basis.

Our reasons for selecting the brazing method may be summarized as follows:

1. <u>Cost Comparison</u>

Our analysis of the cost of electroforming and brazing indicates that the cost of electroforming would be approximately 9 percent higher than the cost for brazing. The electroforming would require approximately 1300 sq ft more floor space than the brazing method.

2. <u>Performance</u>

A review of the technical data indicates that within the accuracy of cold test measurements the electroformed and brazed sections should give equivalent performance.

3. <u>Flexibility</u>

The electroforming method of fabrication has less flexibility, from a scheduling point of view, than the brazing method. It appears unrealistic to plan on producing more electroformed pipes than the capacity of the initial installation, which would yield two or three 10-ft sections per day. Using the brazing method, we have the possibility of producing up to six sections per day on a three-shift basis. Accordingly, the brazing method provides greater flexibility and, hence, more assurance in scheduling.

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* M Report No. 298, April 1962, Stanford University.

** SLAC Report No. 1, July 1962, Stanford University.

By the time the electroforming work ceased SLAC's engineers were able to plate on 3/8 inch of copper in three days, while the normal rate in commercial practice is .001 inch per hour.

In the final design, the sections are made from disks and cylinders, the cylinders cut from pipe, as shown in Exhibit 5, rather than the drawn or forged cups considered earlier. The disk and cylinder construction is less expensive. During fabrication of a section disks and cylinder are lined up in a granite V-block. Then the stainless steel mandrel is inserted and the assembly placed vertically in the furnace. Starting at the top, a ring-burner travels down the section at 10 cm/minute. A copper-silver eutectic brazing alloy with a flow point of 1436°F is used. The entire process, including machining of the rings and disks, was carried out at SLAC because no contractors capable of manufacturing the sections to the required tolerances could be found. Each section cost $3,000 to $3,500.

An electroformed accelerator section (before etching) is shown in Exhibit 6. The problem then facing Al Lisin was to devise a means of holding the temperature of similar brazed sections constant within the limits dictated by the dimensional stability requirements. All permanent components of the accelerator were expected to function for ten years without maintenance or repair.

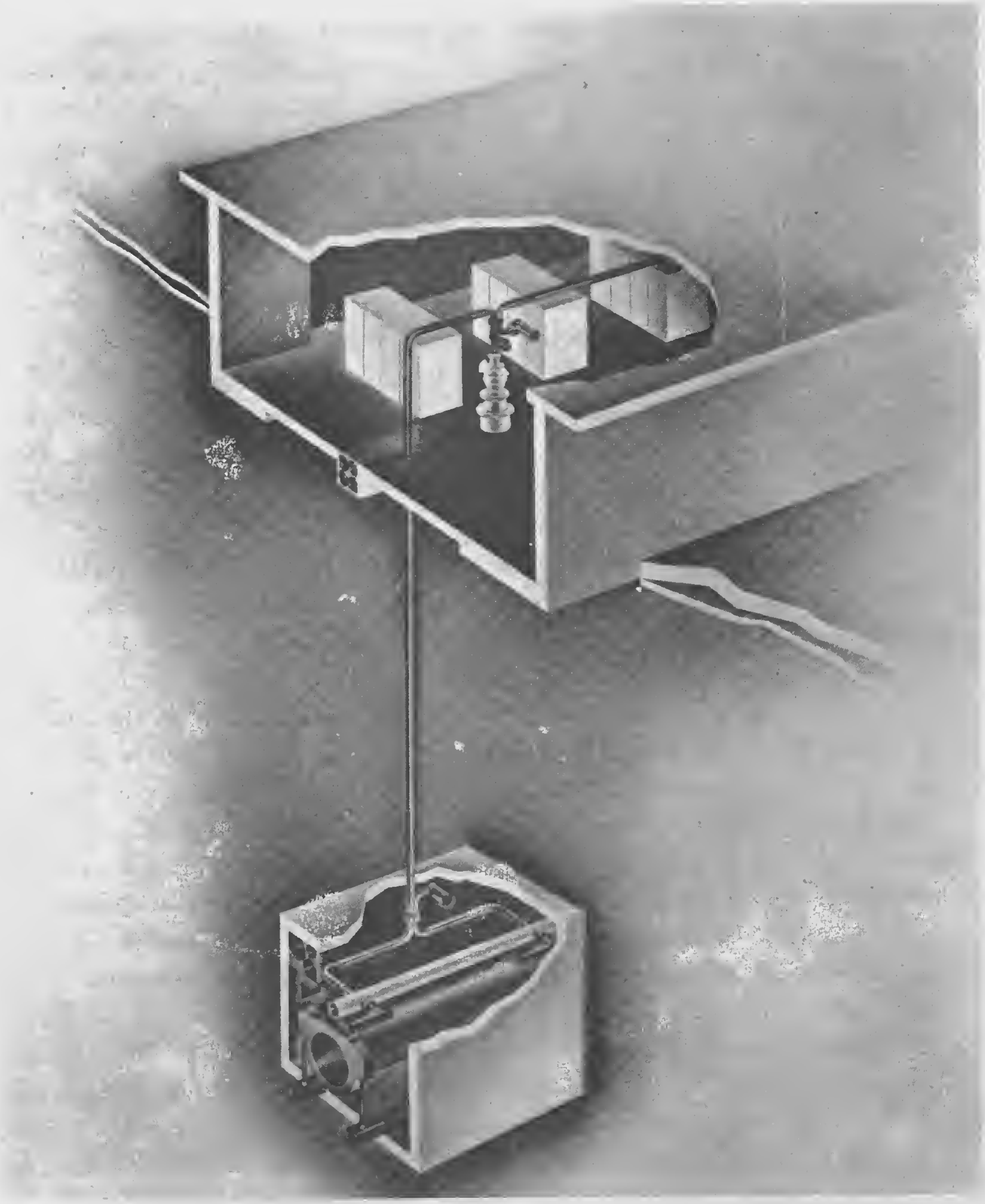


Exhibit 1: Klystron Gallery with Accelerator Itself 25 Feet Below.

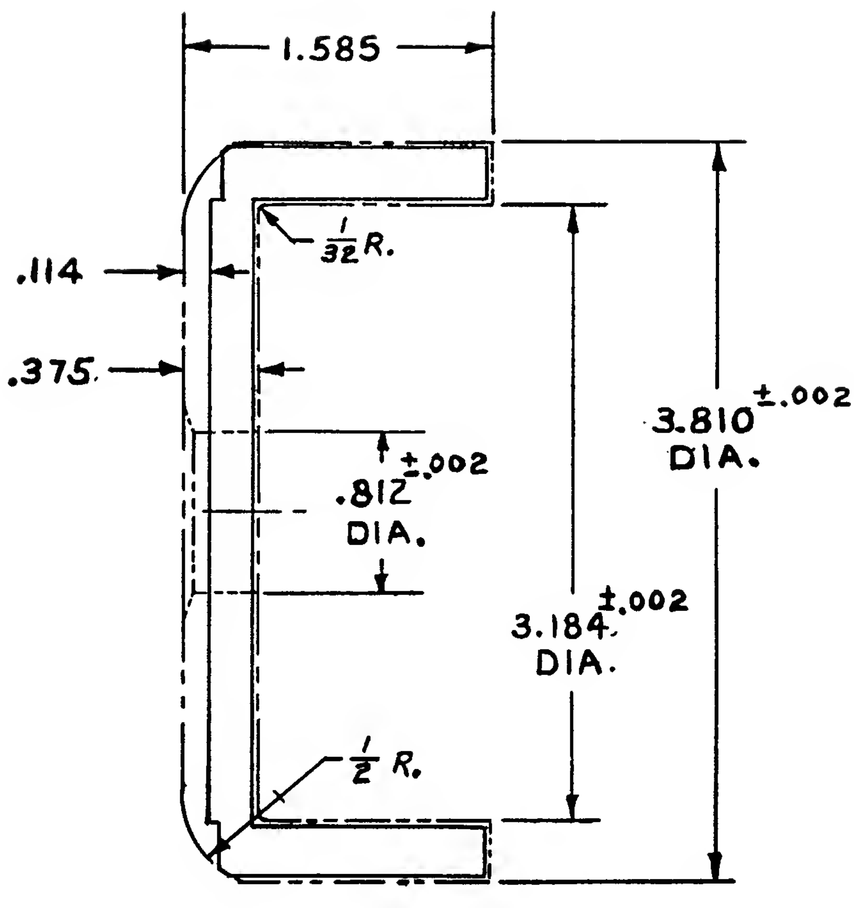


FIG. 2--Blank cup after drawing (2π/3 mode).

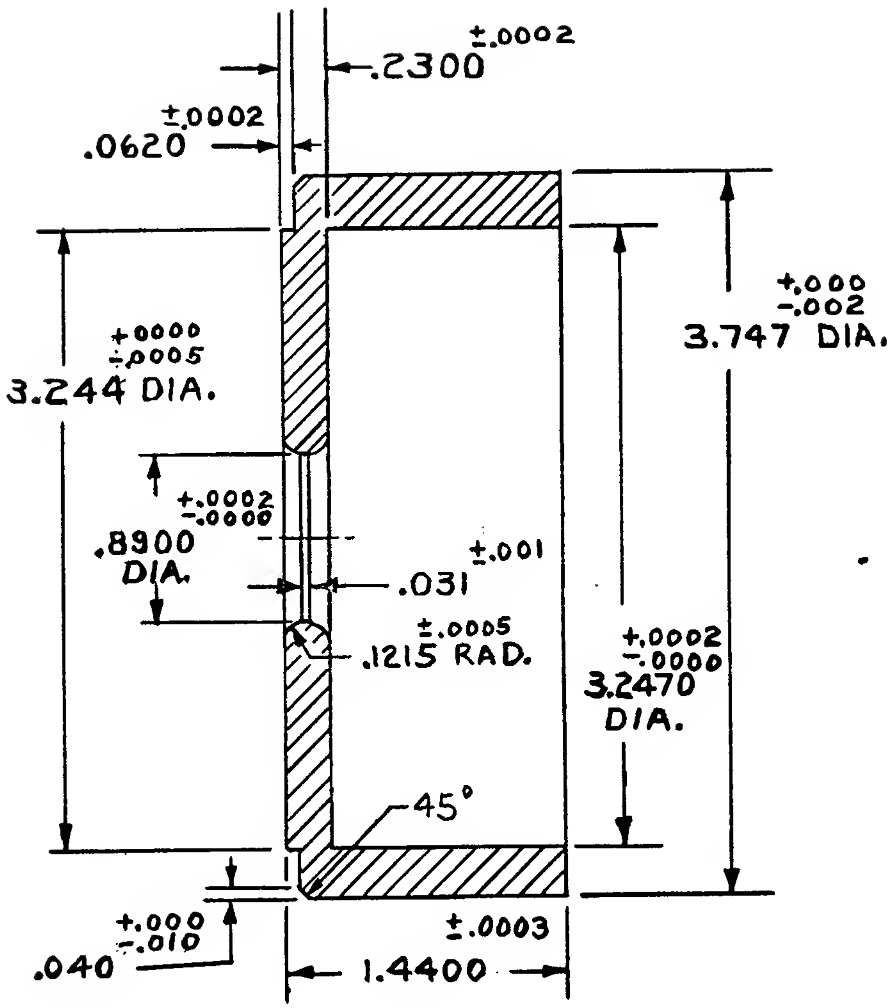


FIG. 3--Cup machined to finished dimensions (2π/3 mode).

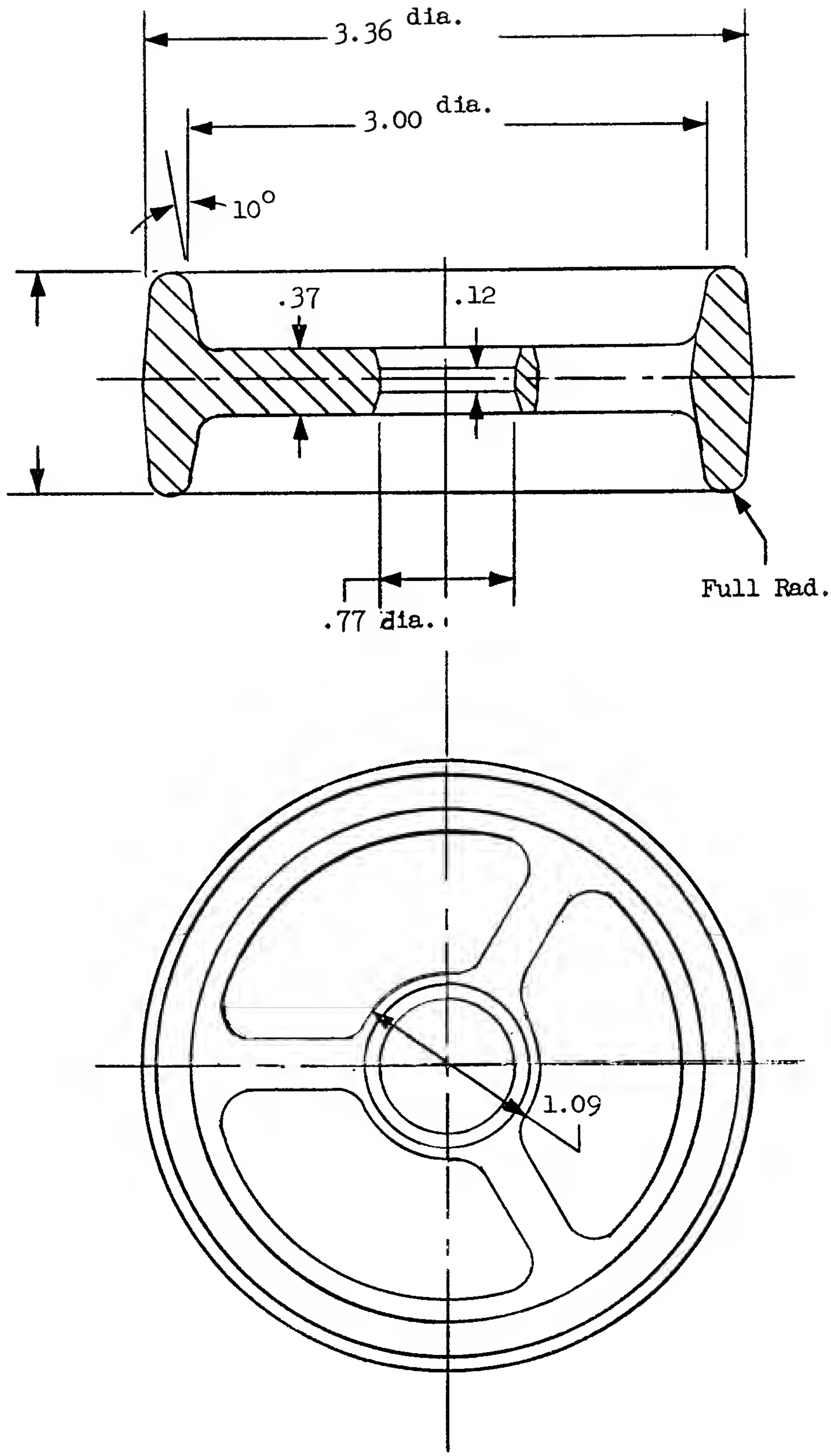
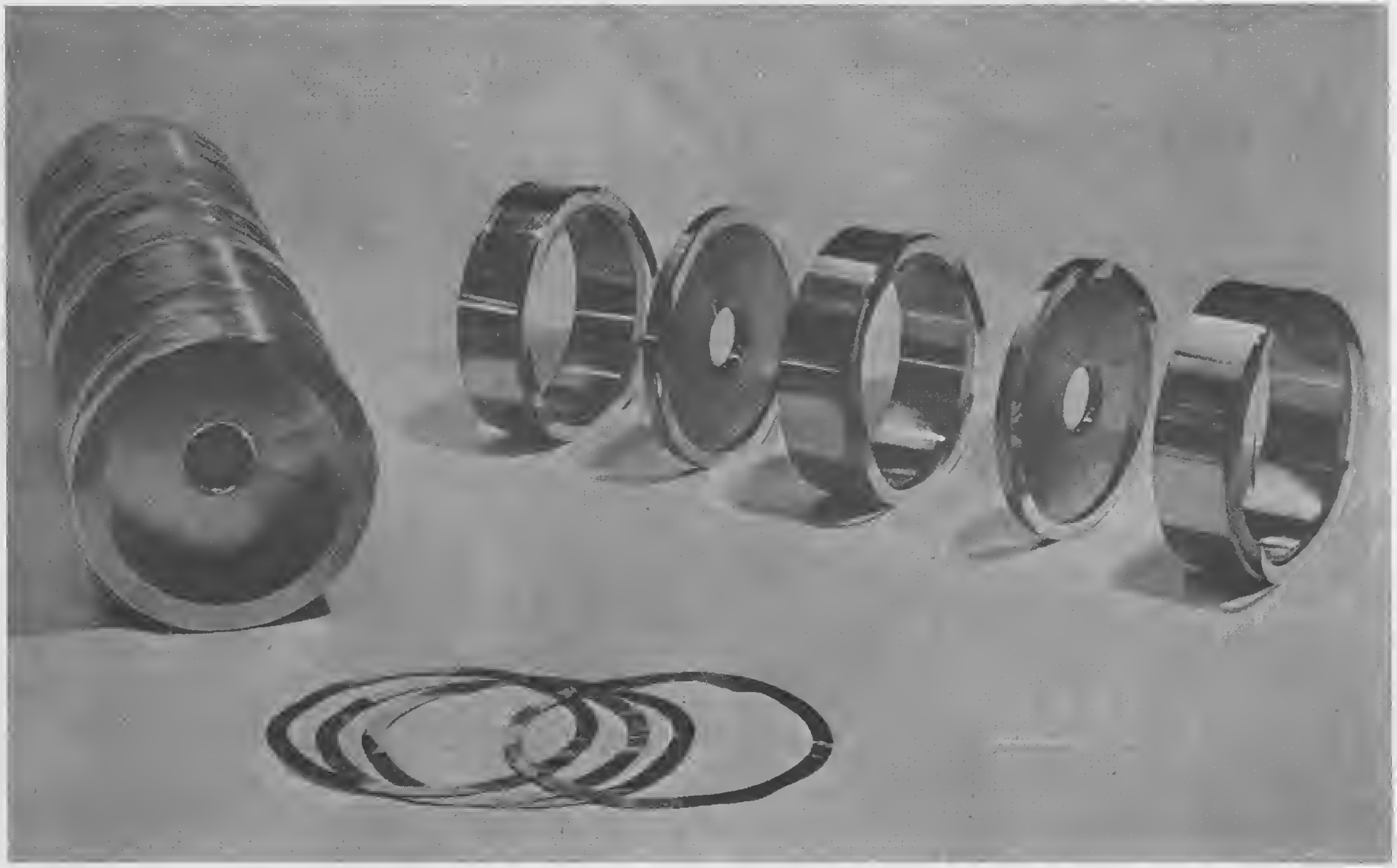


Exhibit 4: Forged Aluminum Spacer.



<u>Exhibit 5:</u> Cylinders, Disks and Washers of Brazing Alloy from which the Beam Pipe is Made.

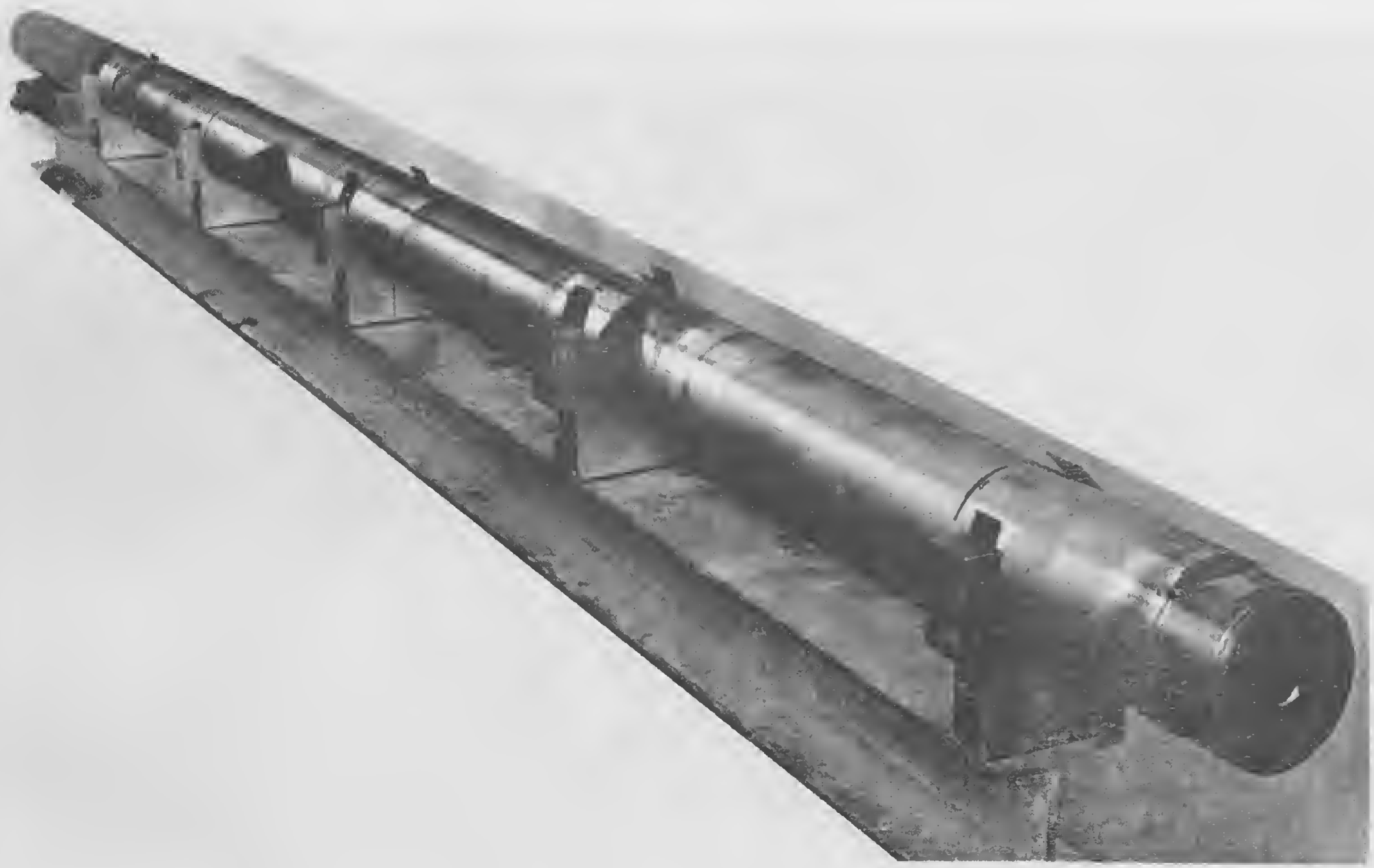


Exhibit 6: Electroformed Accelerator Section (Before Etching).

STANFORD LINEAR ACCELERATOR CENTER II (B)

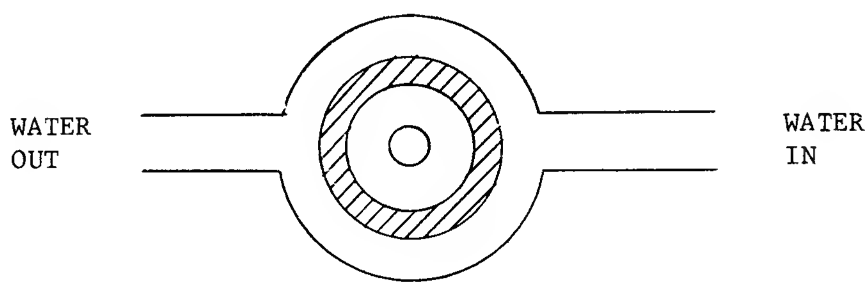
Beam Pipe Cooling System

Four basically different liquid cooling systems were considered by Al Lisin and others during the course of the SLAC two mile accelerator (sometimes called the M accelerator) project.

Crossflow

Al said, "When I joined SLAC in 1961, the talk was of a water jacket. I suspect that they'd looked at ways of air cooling the pipe before and quickly found there's too much heat for that. In 1960, before I was there, they looked at a crossflow water jacket made of sheet metal."

A cross-section of a pipe with a crossflow water jacket is sketched below:



Al continued, "This jacket would have to be put on after the accelerator was tuned, in which case tuning could not be accomplished at operating temperature, or else they would have to find some way of tuning with it in place, but at this time they didn't even know how the sections would be made. I think the more serious problem was that with the short flow length and large area too much water would be needed. Economics quickly rears its head. The higher the flow rate for any cooling system, the more the piping, the pumps and the cooling towers cost."

The 1 July to 30 September 1961, Status Report* stated:

> A small scale test has been planned to determine the required number of water jets for the crossflow water-jacket design for the constant-gradient section. The test will be carried out with a simplified copper model of the accelerator section. The model is being made by putting copper ribs, simulating copper disks, on one side of the plate. On the other side, the water chamber is formed by a second plate that has an experimentally variable spacing. The heat input to the accelerator will be simulated by strip heaters attached to the rib side of the plate. The objective of this experiment is to provide a uniform-heat input to the under side of a copper plate and to measure the temperature distribution on the other side of the copper when water is flowing across the surface under varying conditions. With the data from this preliminary experiment it will be possible to make a test set-up with an actual short section of accelerator to check the results of the data from the sample experiment.

Axial Flow Water Jacket

Full annular jackets in which water or some other coolant would enter at one end of each ten foot section and leave at the other were also under consideration during 1960. A proposed design was described in the 1 January 1960 - 30 June 1960 Status Report (M.L. No. 741) as follows:

> It is desirable to assemble the accelerator sections completely before installing the water jacket. This includes having the end flanges and waveguide couplers brazed on.
>
> In addition to containing the cooling water, the water jacket assembly should also be a structural member capable of supporting the accelerator tube and keeping it straight.
>
> Several designs have been considered using a full length tube slipped over the accelerator section from one end. The tube would have sufficient strength and rigidity to support the accelerator sections. Inside baffles between the jacket and the accelerator pipe would support the pipe and keep it in alignment.

*M.L. Report No. 859, M Report No. 280, October 1961, Stanford University

There are some disadvantages to this type of construction. The jacket would have to fit very closely on the accelerator pipe, and hence would be difficult to slide over the pipe without introducing excessive forces that might deform the soft copper accelerator pipe. Also, the end flange and wave guide coupler would have to be brazed on after the jacket has been installed. This would necessitate tuning the accelerator cavities with the water jacket in place.

A design of a combination supporting structure and water jacket, shown in Figure 5*, shows promise of overcoming most of the disadvantages of other designs that have been considered.

In this design, the accelerator pipe is completely assembled, including the end flanges, waveguide couplers and cooling water inlet and outlet connections.

With the accelerator tube in a vertical position, the angle iron support frame, with the lower half of the water jacket in place, is placed against the table, and clamped to it. The end flanges are clamped to the support saddles. This sub-assembly can now be laid down horizontally for completion of the assembly work.

The upper half of the jacket is then installed and clamped to the angle support with a soft copper wire gasket between the longitudinal flanges of the upper and lower halves. The end flanges of the jacket are heliarc welded to the flanges on the water inlet and outlet connections.

Two complete support frame and water jacket assemblies have been ordered for testing on the new accelerator sections for Mark IV. They should be ready for installation in the early part of August.

The Mark IV was a 20 foot accelerator located on the Stanford campus. The Mark IV accelerator was used by the School of Medicine before being turned over to SLAC. The two sections were built and shortly after Al joined the project were tried on the Mark IV. He said, "They seemed to work very well as far as temperature control went, but they gave a lot of trouble during fabrication. It was very hard to control the gap between the stainless steel jacket and the pipe, and we couldn't get a leak-tight seal at the copper wires. I think they ended up putting radiator stop-leak in it. It was also hard to join two sections like this together, and they cost a lot to build."

*Exhibit 1.

The 1 July to 30 September 1960 Status Report* described another jacket:

> An alternate water jacket, shown in Figure 3** has been ordered for delivery in October. This jacket consists of a full length tube 4-1/2 in.O.D. x .062 in.wall. The tube has been dimpled with circumferential rows of evenly spaced 7/16 in. diameter dimples in line with the cavities in the accelerator tube.
>
> The dimples serve several purposes: they support the accelerator tube centrally in the jacket; they create increased turbulance in the cooling water; and they make it possible to tune the accelerator cavities with the jacket in place.
>
> This jacket is designed to utilize the same water inlet and outlet headers as are used with the split jacket.
>
> A stainless-steel dummy accelerator tube has been fabricated to be used for trial assembly of the water jackets. This assembly will also be used to check the copper-wire-gasketed joint in the split jacket, and to determine the water-pressure drop through the jacket.

The tuning operation referred to would be an indentation of the walls of each cavity.

Reduced Pressure Boiling

Al said, "By putting the whole pipe beneath the surface of a boiling liquid you can control the temperature by controlling pressure above the liquid. They were then thinking of copper temperatures in the range of 80° to 130°F. You can hold the temperature very closely by good control of the saturation pressure."

The reduced pressure boiling idea is sketched in Exhibit 3. The condenser above the pipe is water cooled and connected through a control valve to a vacuum pump. The pressure in the jacket is sensed and the signal fed to a control system which regulates the valve to hold a constant set pressure. Experimental sections were tested during the winter of 1961-62. Al explained that the range of temperatures considered represented compromises, as a low temperature is desirable to minimize the resistivity of the copper and cut the rf losses, while higher temperatures allow smaller heat transfer areas and lower manufacturing costs.

Cooling Tubes

Al said, "The idea of spacing a number of small longitudinal cooling tubes around the pipe came when I started to worry more about how we

* M.L. Report No. 770, M Report No. 232, November 1960, Stanford University
** Exhibit 2.

would be able to dent or dimple the pipes for tuning. If we wanted to indent at several points around each of the 86 cavities, and do this with water running through a jacket, then obviously one of the simplest ways would be to just use parallel tubes, leaving the surface of the pipe exposed between them. If some other kind of jacket were to be brazed on after tuning, the high temperature could destroy the adjustments.

If we decided to use a constant structure pipe, the increase in cooling water temperature along with pipe would balance the decrease in heat generation while for a constant gradient pipe we could run water in opposite directions in adjacent tubes."

The first mention of the cooling tube design came in the 1 October to 31 December 1961 Status Report*;

> Another promising water-jacket concept is currently being investigated. In this cooling method, 16 one-half inch o.d. tubes are brazed directly onto the surface of the accelerator. Water flows in one direction through all tubes for constant-attenuation sections, and in opposite directions in adjacent tubes for constant-gradient sections. Test samples are being made to determine the possibility of brazing the tubes in a single operation in the vertical brazing furnace. A test system to evaluate this concept has been designed and is currently being fabricated.

Al explained, "We tried 16 tubes because it seemed as if it would be a good idea to cover as much of the surface as possible. The tubes were 1/2 inch o.d., 3/8 inch i.d. commercial copper water pipe. At this point I'd only made a few very rough calculations, and the purpose of the test was just to see how well we could hold the temperatures. Some of the other advantages of the tubes are pretty obvious -- fabrication is simple compared to a sheet metal jacket and dimensions can be held closely, which helps give uniform temperatures. Also there's little chance of leakage, since connections to small round tubes are easy to make. In parallel with the experimental work I was going to look at the size and number of tubes it would be best to use."

The test section built to be tried on the Mark IV is shown in Exhibit 4.

* M.L. Report No. 882, M Report No. 291, January 1962, Stanford University

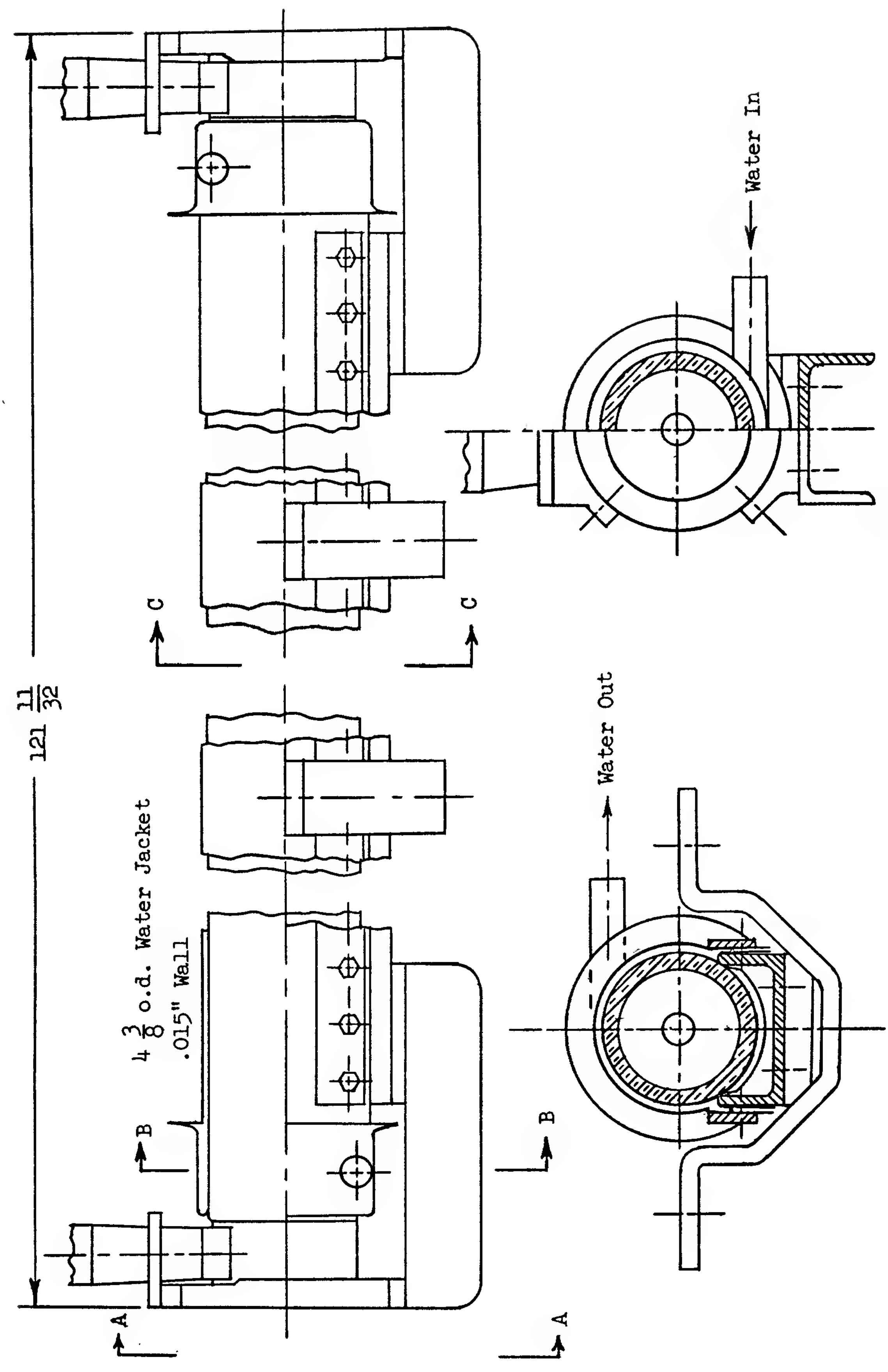


FIG. 5--Possible assembly of water jacket and supporting section.

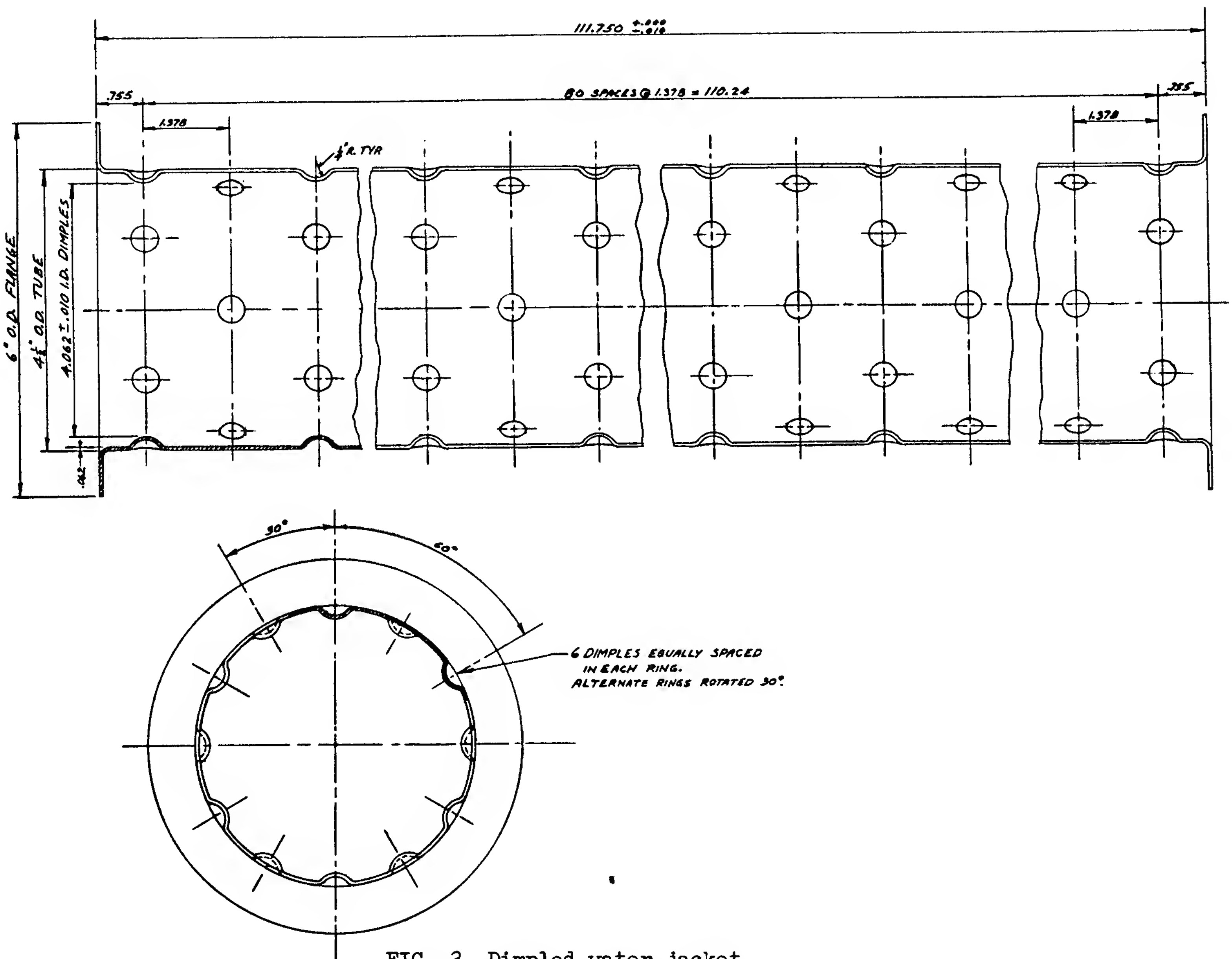


FIG. 3--Dimpled water jacket.

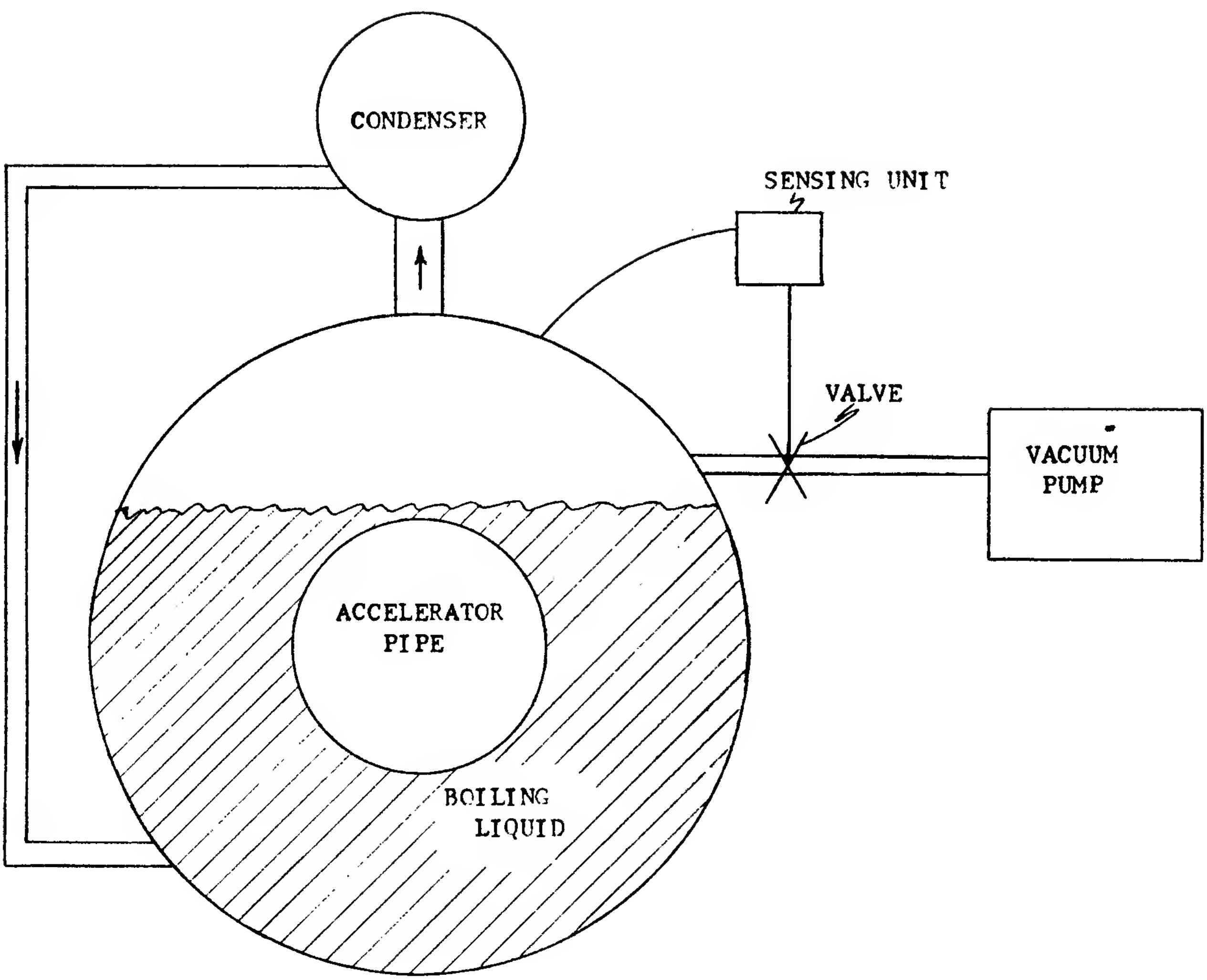


Exhibit 3: Schematic Diagram of a Reduced Pressure Boiling System.

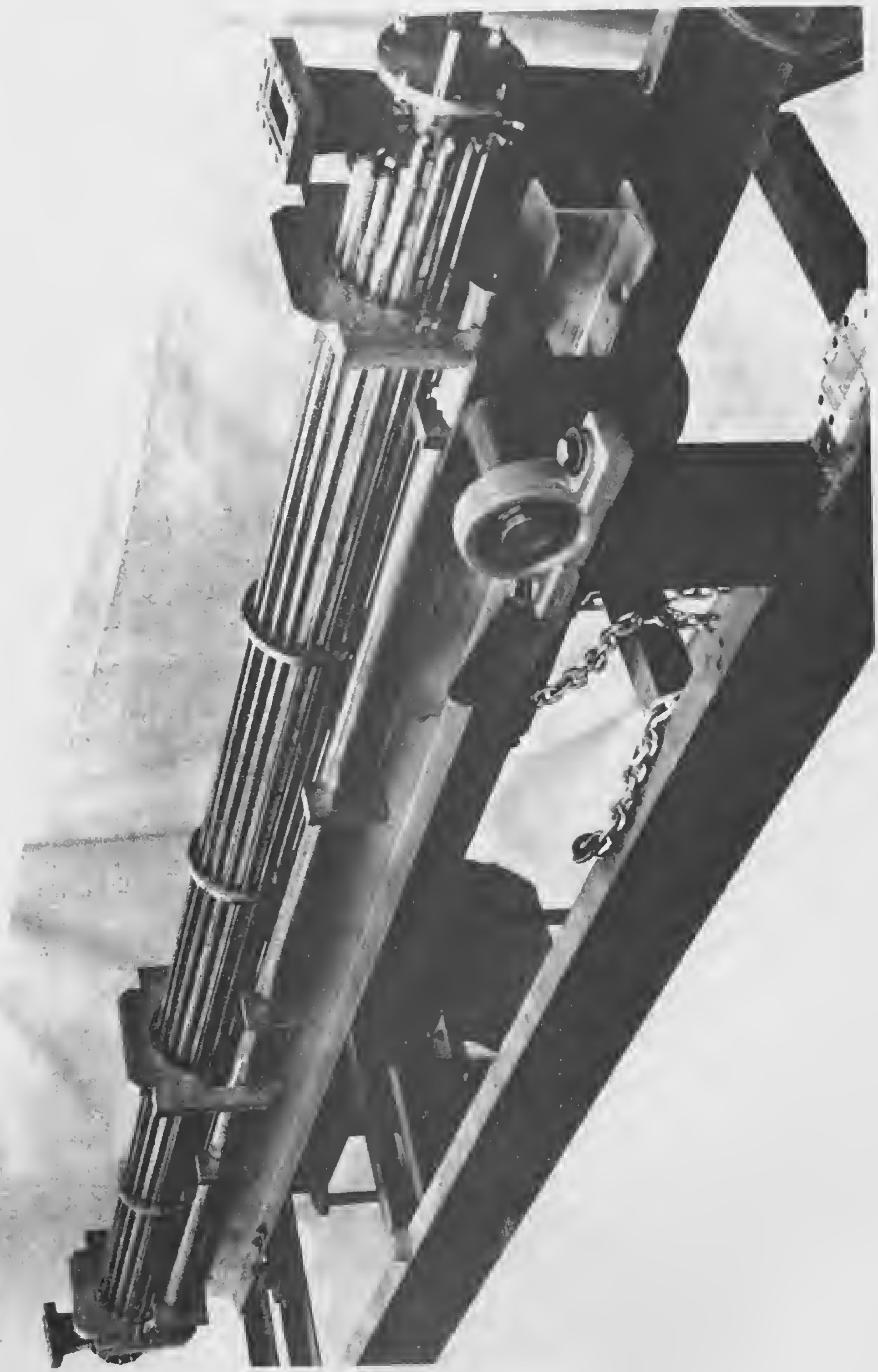


Exhibit 4: Test Section with 16 Longitudinal Cooling Tubes.

STANFORD LINEAR ACCELERATOR CENTER II (C)

Beam Pipe Cooling System

The January 1962 Status Report commented on the cross-flow cooling arrangement as follows: "The small-scale, cross-flow-test mockup was set up and tests performed. The tests have shown conclusively that unless an extremely large amount of water is used, the cross-flow method of cooling the accelerator cannot meet the ± 1/4°F temperature-control requirement."

The same report mentioned that a further development of the axial flow full water jacket was being evaluated but that it probably would not be used on the M accelerator because of fabricating and installation difficulties. Al commented, "Actually, a sheet metal water jacket in theory is a very good way to cool the pipe, although we had to drop the idea because it's so hard to hold the annular dimensions constant. For a uniform structure accelerator the size of the annulus and the flow rate can be chosen to give a constant pipe temperature, with the water entering at the high heat generation end and flowing toward the low heat generation end. You can show analytically that there is a single flow rate for any geometry that gives an axially uniform accelerator temperature. Then the temperature can be held constant regardless of the RF power level by varying the water inlet temperature."(Exhibit 2).

"Things are a little more complicated for a constant gradient structure. With uniform heat generation you can taper the annular space so that it decreases towards the water outlet end. At the large end, velocity is low and the film conductance is also low. This gives a large water-to-pipe temperature difference. Towards the other end, both water temperature and velocity increase, the conductance increases, and the temperature difference is smaller. The other way to take care of the constant gradient case is to vary the flow rate along the section. As the water temperature increased, the flow rate would also be increased. To do this we would have had to use a double water jacket. Either way, the fabrication problems would have been enormous for the constant gradient alternative."

The April 1962 Status Report (M No. 298) said:

> The reduced pressure boiling test system operation continued. Both test results and analysis have indicated that reduced pressure boiling cannot be used to maintain a uniform accelerator pipe temperature if the pipe is submerged. The difference in the

depth of immersion between the top and bottom surfaces of the accelerator pipe (4 in.) leads to a temperature difference of 5°F from top to bottom. Other methods of applying the water to the surface, such as spraying, resulted in nonuniform spraying coverage with a resultant nonuniform wall temperature. The reduced pressure boiling cooling method has, therefore, been dropped from further consideration.

Al commented, "The 5° temperature difference would lead to severe bowing of the beam pipe. When we tried keeping the water level below the pipe and spraying water on we ran into problems with particles and impurities in the water plugging or corroding the nozzles; this was with a nozzle velocity of 40 to 50 feet per second. An electrically heated test section was made with a window in it, and we could see that we weren't getting uniform coverage on the pipe. We also tried using wicks to get the liquid to the pipe surface, but again got poor coverage. Although the water temperature stayed constant, dry areas on the pipe were at high temperatures while the temperatures were too low where the water dripped off. When we went to very small nozzles to give a finer mist we ran into the plugging problems. Then we started looking at fluids with a higher saturation pressure and a lower rate of change of saturation temperature with pressure than water. This would allow higher pressures and less of a temperature change over a submerged pipe. Only a few organic fluids would have worked, but they all tend to decompose in the radiation."

Concerning the cooling tube work, the April 1962 Status Report said:

2. <u>Longitudinal Cooling Tube Design</u>

The longitudinal cooling tube concept in which many parallel cooling water tubes are attached to the outside surface of the accelerator pipe was investigated in greater detail. Analytical work has indicated that, for constant-attenuation operation, water flowing in parallel through these tubes would be capable of maintaining a constant circumferential average temperature along the full length of the accelerator pipe. The same cooling tube arrangement with water flowing in opposite directions in alternate tubes can be used for cooling the accelerator pipe under constant-gradient operating conditions. Analysis indicates that, with cooling tubes of constant cross section and with an appropriate tube size and water flow rate, the circumferential average wall temperature can be held constant all along the accelerator pipe within about 0.6°F. The possibility of tapering the tubes in a manner that would vary the heat transfer characteristics along the tube and thereby allow the attainment of a more nearly constant temperature is also being studied.

A test system has been fabricated and set up for evaluation of the longitudinal cooling tube concept. The heat input system has been debugged and is now operating properly; the constant temperature water supply system has been checked out; and the thermocouples have been calibrated and installed. Preliminary data indicate that the constant-diameter longitudinal tube arrangement is probably adequate for maintaining a constant

accelerator temperature at constant-gradient operation for the Mark IV accelerator and for Stage I power levels. Further testing with straight and with tapered tubes is planned. Figure 4* shows the test system prior to completion of insulation around the dummy accelerator section.

3. Water Temperature Control System

A water temperature control system capable of supplying water at a constant temperature to within ± .2°F, at flow rates up to 20 gpm, and with a 35 psi head, was designed and built for use in the cooling system tests. The system is a closed circulating loop built entirely of stainless steel and non-ferrous materials to maintain the water in a state of high purity. Operation to date indicates that the system will have negligible corrosion. A series of test runs at power levels from 1 kw to 20 kw has been completed, and the temperature of the water has been shown to be constant with a maximum deviation of ± 0.2°F over the entire range. Figure 4 shows the water temperature control system at the right.

In the July 1962 report (SLAC-1) it was stated that, "A minimum of 13 gpm per 10-ft accelerator section will be provided. The accelerator tube metal temperature will be held at 113°F (45°C) by varying the temperature of the water supplied."

Al said, "The 13 gpm and the 113° temperature both came out of committee meetings. The temperature was a compromise between efficiency and cost; we were still considering flow rates greater than 13 gpm but we had decided the flow rate would be held constant. This is because there's only a limited range of water velocities in the tubes that give turbulent flow without excessive corrosion and erosion. I checked the literature and 7 feet per second seemed to be about the upper velocity limit for water flowing through copper tubing. The range is small enough so we couldn't control temperature by changing flow rate alone over the whole range of RF power, so there's no reason to vary flow at all."

A graph Al had made in January relating some of the variables is shown in Exhibit 3.

By the time the test work was well underway, in August 1962, it had been decided that the M accelerator would be of the constant gradient type. A paper**by W.K.H. Panofsky, the Director of SLAC, stated the reasons for the choice as follows:

* Exhibit 1

** PROGRESS REPORT ON THE STANFORD TWO-MILE ACCELERATOR, presented at the 1963 International Conference on High Enery Accelerators, August 1963, printed in SLAC-18.

The accelerator proper is fabricated in 10-foot "sections" of the constant gradient type. By this, we mean a structure designed to propagate electromagnetic waves in a longitudinal electric circularly symmetric mode with phase velocity c, but with group velocity v_g varying ($0.020\ c \geq v_g \geq 0.007\ c$) such that the electric field will build up to an essentially constant value. This is achieved by varying the cavity dimensions (three disks per wavelength have been chosen to optimize the shunt impedance) for each of the 86 cavities constituting the section. The choice of the "constant gradient" structure has considerable advantages, the most important being a lower peak electric field strength for a given energy gain and also less problems with multipactoring and "beam breakup" due to unwanted modes.

After the August decision, all further experimental and analytic work concerned only constant gradient sections. Drawings of the disk and cylinder pieces which make up each section appear in Exhibits 4 and 5. Slightly different cylinders go on each end of the section, thus only 84 are listed on the drawing of Exhibit 4.

Also from the July 1962 report.

Several tests of the longitudinal cooling-tube arrangement were made using an electrical heater in a mock-up of an accelerator pipe. Individual temperature measurements were not always consistent, but averages of many readings showed that, at full power conditions along the accelerator mock-up, temperatures were held at 0.2°F. The problem of obtaining accurate temperature measurements has been studied in some detail. A technique for making thermocouples having the required degree of accuracy has been developed. New thermocouples accurate to within 0.1°F are currently being installed on the accelerator mock-up

Al commented, "The first dummy section was just a ten foot copper pipe, with a four inch O.D and 3/8 inch wall and the 16 tubes soft-soldered on. We had to solder them because we didn't have a furnace big enough to handle a ten foot length of beam pipe and you can't get it hot enough for brazing with a torch. When we tried to use the flame furnace and ring burner that the pipes are brazed with, we burned up the tubes before the beam pipe even got hot."

"We put a moly rod in the center of the pipe, evacuated the pipe to keep the moly from burning up, and poured up to 15 kw of power into the rod while we measured temperatures on the pipe surface. The temperature variations were as much as 9°F. We thought a good part of this was due to the poor thermal conductivity of the solder and also because we didn't have a uniform solder joint, so then we made up another section, soldering more carefully We still had trouble because of non-uniform heat input and our thermocouple readings were subject to a lot of error. We decided to simplify the setup and brazed a single tube to a copper bar with a

heater under the bar. I had started to worry about entrance effects in the tubes, since there would have to be a sharp bend where the water entered the tube from the manifold and the conductance can vary quite a bit as the boundary layer builds up. We put thermocouples on the bar in the entry region to try to check on this. Overall, I thought this test was inconclusive because we were still having problems getting good temperature readings, but the results did show that entrance effects were smaller than would be predicted from data in the literature."

Soon afterwards a test stand was finished in which beam pipes could be tested with an RF power input such as they would experience in the accelerator. This test stand was also used to determine the effect of pipe temperature variations on RF phase shift.

Al continued, "About this time we tested a section with only eight tubes in addition to the one with 16. By then it seemed that the tuning would probably be done by indenting the walls of each cavity at four equally spaced points. This meant the number of tubes could be any multiple of four. It seemed obvious that if only four were used the peripheral temperature variation would be too great. Twenty or more tubes would have made the indenting difficult, and 8, 12, or 16 seemed feasible. We did the testing on sections with 8 and 16 tubes because we could interpolate to get results for 12 tubes."

Sections with both 8 and 16 tubes soldered in place were tested in the RF test stand. An eight tube test section with different manifolding from the section shown in Exhibit 1 appears in Exhibit 6. Al said, "The tests showed little difference in temperatures, as near as we could tell, but we were never able to measure temperatures with very good accuracy. I think we finally got down to plus or minus .3 or .4°F after a long struggle. At first we didn't know if we were getting bad temperature readings or if we actually had a severely non-uniform temperature distribution, but when we put about 50 thermocouples on one of the pipes, it seemed that the general trend was toward uniform temperatures, with some erratic readings from thermocouples thrown in. Part of the trouble was that we had kilovolts of RF noise imposed on millivolts of thermocouple output. It was also difficult to fix the thermocouples to the pipe surface. Actually, we didn't really know where we wanted to measure temperatures. To keep the cavity dimensions constant, the volume average temperature of the pipe must be kept constant. But all we could measure were a few local temperatures on the surface, while to get the volume average you have to integrate a three-dimensional temperature distribution. This posed a real problem for a control system -- where on the accelerator section could we sense a temperature that would be representative of the volume average?"

"We wanted to use a lot of thermocouples to measure both the axial and circumferential temperature variation, and not only between the cooling tubes but also around them. But when we stuck them to the surface, the variations in contact resistance introduced too much error.

On electroformed accelerator sections we were able to pierce the outer layer of copper and pry it up to make a cavity for the couple. This is possible because the electroplated copper is deposited in layers. Then we peened down the cavity over the wires. But this wouldn't work on brazed sections. With these we tried drilling holes and peening the metal down around them. We also tried contact type devices that are pushed against the surface with a handle, but these gave varying readings depending on how hard you pushed."

The method that Al finally felt gave the most reliable results used a copper pin and a constantan pin, each made from 14 gage wire, set into holes drilled 1/4 inch apart in the wall. The holes were 1/8 inch deep and the pins were pressed in. After they were inserted the adjacent surface was peened down to stake them in. The temperatures were read with a Leeds and Northrup "Speedomax H" indicator having a guaranteed repeatability of .15°F over the range 50 to 150°F. On sections instrumented in this manner with soldered cooling tubes the circumferentially averaged temperatures fell in a range of 0.9°F along the ten foot length for a range of RF power inputs averaging up to 17 kw. The maximum temperatures were in the middle of the section.

Al said, "The tests on accelerator sections couldn't tell us the best size and number of tubes to use; for this I had to rely on analysis. Around the end of 1962 I was also worrying about the control system and how to manifold the sections. With eight connections to each of two manifolds at both ends of the section, things were pretty complicated. We used circular tubes around the pipe, but the connections were hard to make and I was worried about getting different flow rates in the different tubes. At one point in the design we thought we would have to squeeze the tubes to adjust the flow rates until they were equal."

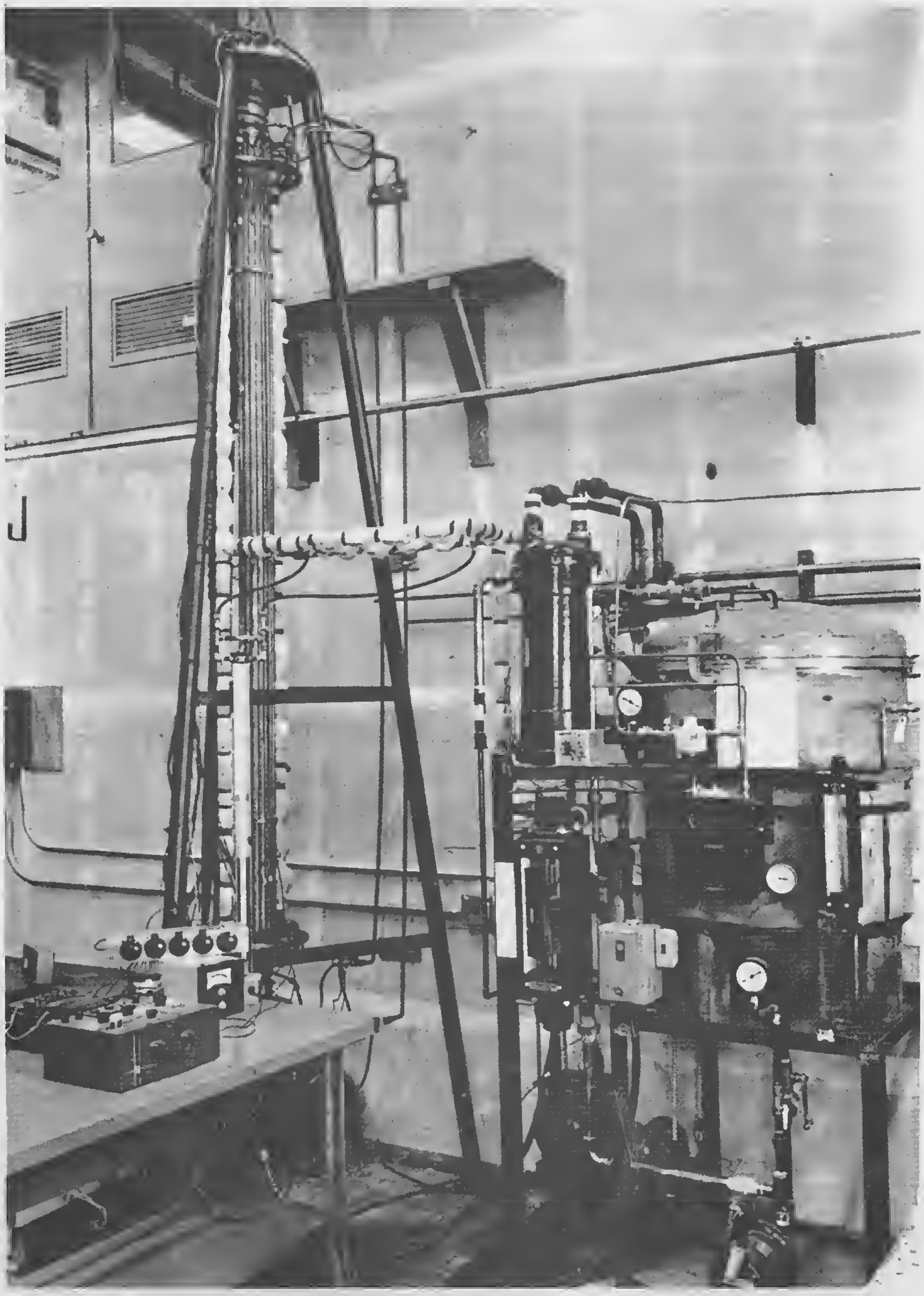


Exhibit 1: Cooling Tube Test Set-Up (Figure 4). Note the Two Manifolds Encircling Each End of the Accelerator Section.

TN-62-23
A. V. Lisin
April 9, 1962

TENTATIVE WATER-JACKET WATER REQUIREMENTS

The proposed method of cooling the accelerator pipe is by longitudinal cooling tubes attached to the outside surface. The tubes would be manifolded so that the flow through all is in one direction for constant-attenuation operation and in opposite directions in alternate tubes for constant-gradient operation. This cooling method appears desirable because it allows the accelerator sections to be tuned after the tubes are attached, because no close tolerance parts are needed, and because, hopefully at least, the tubes can be attached by brazing in a flame furnace.

Constant-Gradient Operation

In constant-gradient operation, the heat generation along each accelerator section is uniform, thus one cannot provide cooling water at one end of the section and remove it at the other and maintain a constant wall temperature without varying the heat transfer characteristics alongs the tube. Since it would be desirable to use tubes with a constant size along their length, the counter-flow arrangement was investigated. In this cooling system, the temperature rise of the accelerator wall under one tube is approximately compensated for by the temperature rise in the opposite direction under the adjacent tube, yielding an average wall temperature which is approximately constant. This cooling method cannot yield an exactly constant accelerator-pipe temperature along the entire length. The ends will always be slightly warmer than the mid-section. However, if one is not limited to any specific number of tubes, tube size, or water flow, a constant average temperature can be approached as closely as is desired.

Analyses indicate that for this cooling method it is desirable to have a large accelerator pipe to water temperature difference compared to the water temperature rise. Since it is desirable to keep the accelerator pipe temperature as low as possible and economics dictates the use of as high a cooling water temperature as possible, we see the need for large water flow rates. Large flow rates mean many or large cooling tubes in order to keep water velocities down (to minimize erosion and pressure drop). There is a practical

limit to how many tubes of a given size can be attached to the accelerator pipe. The use of sixteen 1/2-in. OD tubes was found to be a practical compromise. A flow of at least 15 gpm would be required to maintain the accelerator pipe temperature within a $\pm$ 1/2°F tolerance. Increased flow decreases the end to middle temperature difference but the improvement decreases exponentially with flow. A lower flow rate could be used if the tubes were tapered to vary their heat transfer characteristics along their length. A lower limit on the water flow rate would then be 12 gpm in order to maintain fully turbulent flow.

Experimental results to date show the longitudinal tube cooling system to be promising, but are not entirely in agreement with analysis. The difference between the average end temperature and the average mid-section temperature was measured to be 0.3°F at 11.2 gpm and increased to 0.5°F at 15.4 gpm at Stage II heat-input conditions. At lower power levels, this effect was much smaller or nonexistent. Since there is some uncertainty in several of the temperature measurements, the tests will be repeated with improved instrumentation at the earliest possible time. In the meantime it can be assumed that the analytical results are valid.

Constant-Attenuation Operation

In constant-attenuation operation, the heat generation decays exponentially along each accelerator section. Thus, with the proper combination of number and size of water tubes, flow rate, and water temperature, it is possible to maintain a constant accelerator-section temperature over its full length.

Analysis indicates the following trends for constant-attenuation operation: the tube size decreases slowly with increasing water flow and tube size increases with increasing system temperature. The following table indicates the relationship between water flow rate, average water temperature, and tube diameter and number.

TN-62-23

April 9, 1962

No. Tubes and Size	Average Water Temperature 80°F	100°F	120°F
14-5/8 in.	--	--	16 gpm
16-1/2 in.	15 gpm	28 gpm	~50 gpm

The minimum flow rate to guarantee fully turbulent flow is about 12 gpm and the upper flow rate limit (from pressure drop and erosion considerations) appears to be between 35 and 40 gpm.

Conclusions

From the previous discussion it appears that almost any flow rate in the range 12 - 35 gpm can be used for either constant-gradient or constant-attenuation operation. However, the range 15 - 20 gpm appears to be the most desirable, the exact flow rate desired being dependent on the operating temperature. There does not appear to be a significant effect of flow rate on the water jacket cost. Within a practical range of tube sizes, the water jacket pressure drop is not limiting, about 1 psi at 15 gpm and about 8 psi at 40 gpm.

It therefore appears that within the above limitations, the cooling water flow rate must be chosen on the basis of the water supply system economics. It should be remembered that the convection heat-transfer process is quite sensitive to flow perturbations and build-up of corrosion or erosion products, and in addition is difficult to predict to the high degree of accuracy required in the present application. Therefore, in choosing a water temperature-control system, its ability to maintain the desired temperature and a constant flow rate should be considered in addition to economic considerations.

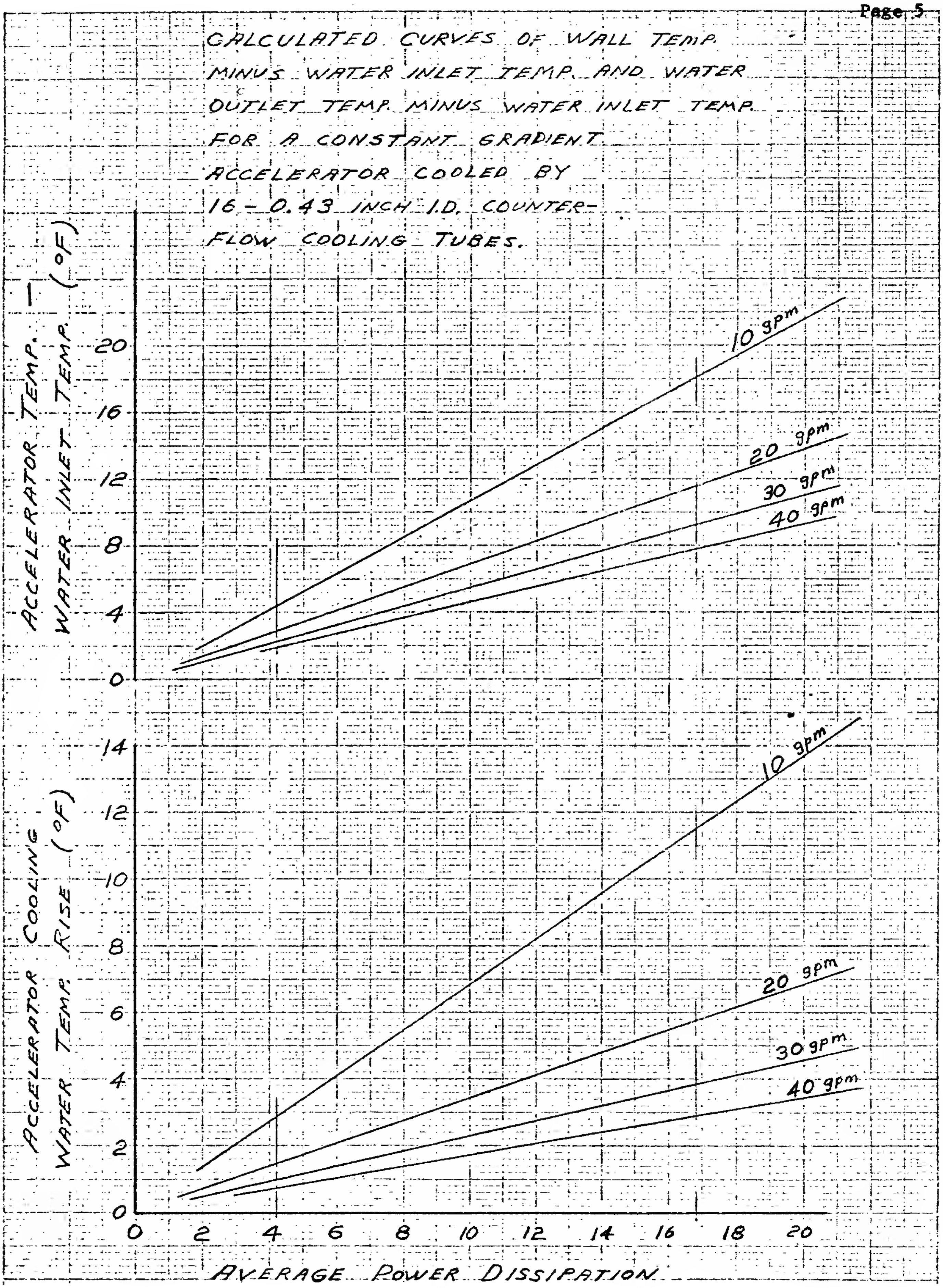
Supplement No. 1
TN-62-23
April 19, 1962

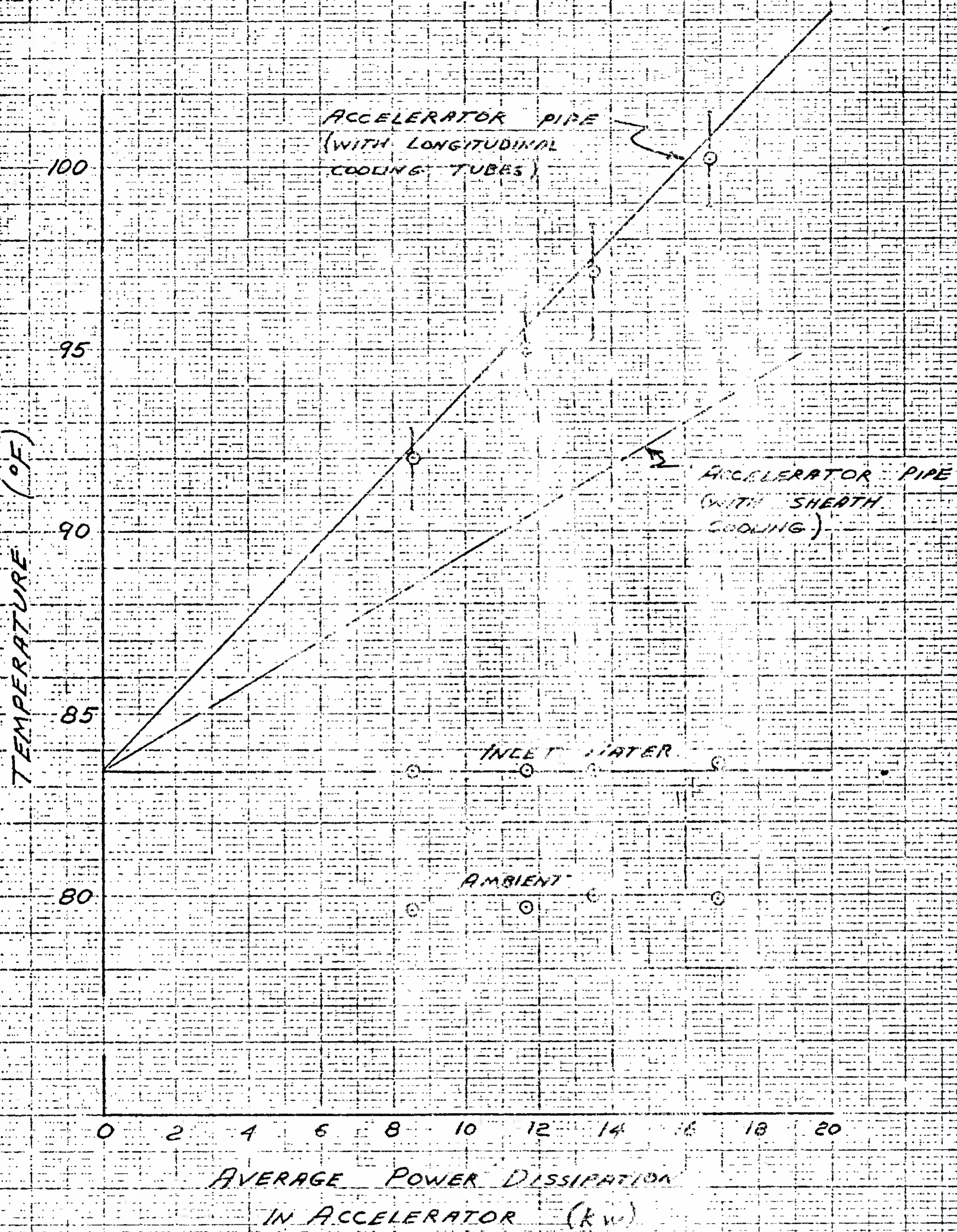
TENTATIVE WATER-JACKET WATER REQUIREMENTS (Supplement No. 1)

Since the subject memo was issued, several questions have arisen concerning the effect of power level and water flow rate on the accelerator pipe temperature. The attached curve shows the accelerator cooling water temperature rise as a function of the heat input and the water flow rate. The curve also shows the difference between the accelerator surface and water inlet temperatures as a function of the heat input and water flow rate. A significant observation which can be made from the accelerator minus water inlet temperature curves is that for a given available inlet water temperature, the accelerator pipe temperature decreases with increasing flow rate. Obviously, it is desirable to keep the accelerator temperature as low as possible, therefore, the higher flow rates are desirable.

The calculated results shown agree with experimental data obtained to date. This can be seen in the second figure where the accelerator pipe temperature is shown as a function of the heat dissipation rate for 83°F inlet water flowing at 10.3 gpm through 16-0.43 inch ID tubes in a counterflow arrangement. The data were taken on 4-18-62 with a constant heat generation rate along a straight piece of pipe simulating an accelerator section and using an electric heater to simulate the RF heat generation. The experimental values of accelerator pipe temperature are the averages of 12 individual readings taken on the pipe surface. The spread of the 12 individual readings is also shown. It should be noted that the test data presented here is preliminary. Future measurements will be made with more accurate temperature sensors, in a pipe with discs and with a more uniform flow distribution through the cooling tubes. Final tests will be performed on accelerator sections with RF heat input.

For comparison with the present longitudinal tube design the temperature of an accelerator pipe with an annular sheath of cooling water (previous water jacket design) is shown by the dashed line. Although the annular water jacket is more desirable from the point of view of temperature, its higher cost, difficulty of installation, and lower reliability are felt to outweigh its temperature advantage. Its temperature advantage becomes insignificant at the higher water flow rate.

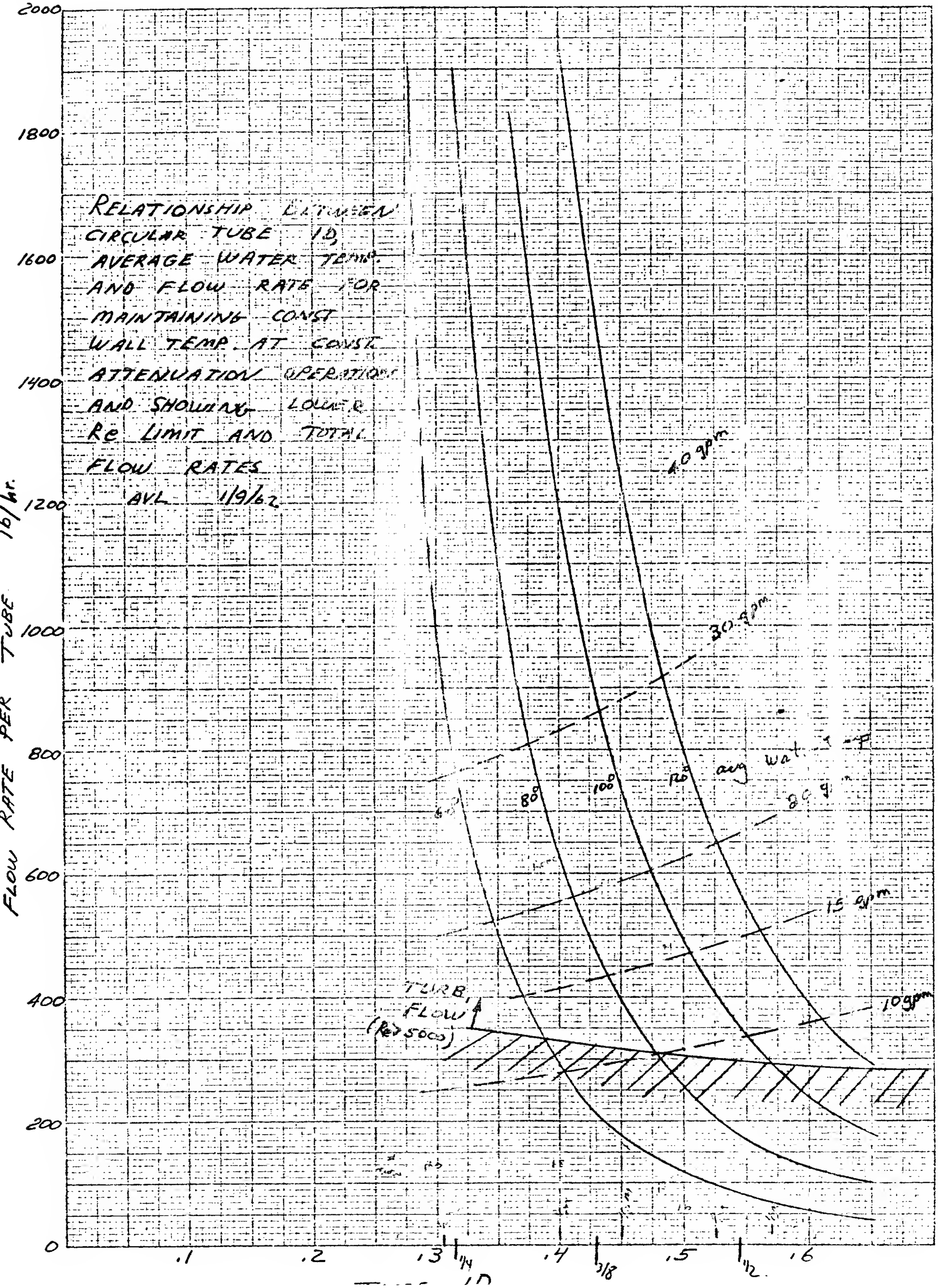

CALCULATED CURVES OF WALL TEMP. MINUS WATER INLET TEMP. AND WATER OUTLET TEMP. MINUS WATER INLET TEMP. FOR A CONSTANT GRADIENT ACCELERATOR COOLED BY 16 - 0.43 INCH I.D. COUNTER-FLOW COOLING TUBES.
ACCELERATOR TEMP. — WATER INLET TEMP. (°F)
0
4
8
12
16
20
10 gpm
20 gpm
30 gpm
40 gpm
ACCELERATOR COOLING WATER TEMP. RISE (°F)
0
2
4
6
8
10
12
14
10 gpm
20 gpm
30 gpm
40 gpm
0
2
4
6
8
10
12
14
16
18
20
AVERAGE POWER DISSIPATION



COMPARISON OF CALCULATED AND MEASURED ACCELERATOR PIPE TEMPERATURE FOR 83.4 °F INLET WATER AND 10.3 gpm FLOW.

AVL
4-18-62

Exhibit 3

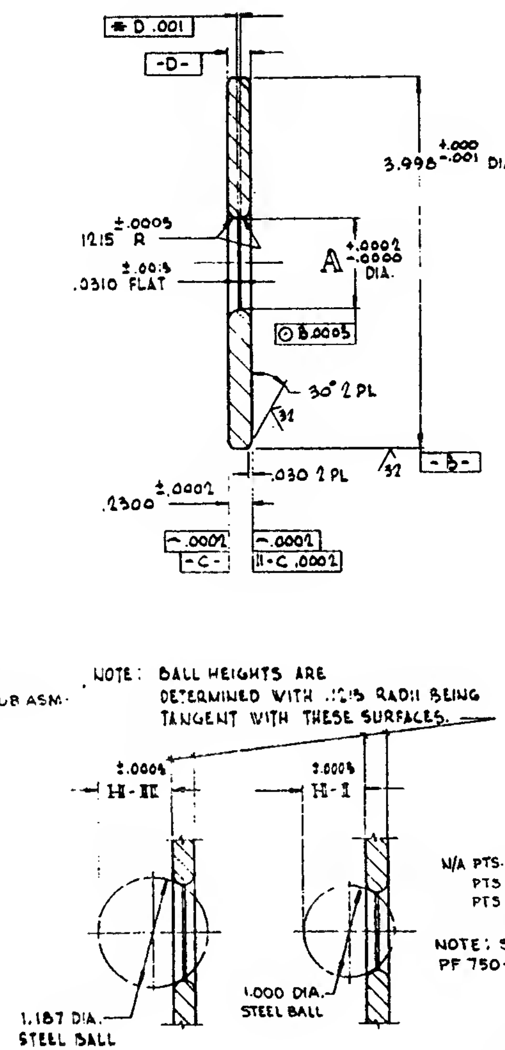

RELATIONSHIP BETWEEN
CIRCULAR TUBE ID,
AVERAGE WATER TEMP.
AND FLOW RATE FOR
MAINTAINING CONST
WALL TEMP. AT CONST
ATTENUATION OPERATIONS
AND SHOWING LOWER
Re LIMIT AND TOTAL
FLOW RATES
AVL 1/9/62
40 gpm
30 gpm
20 gpm
15 gpm
10 gpm
60°
80°
100°
120° avg wat. T °F
TURB. FLOW
(Re > 5000)
FLOW RATE PER TUBE lb/hr.
2000
1800
1600
1400
1200
1000
800
600
400
200
0
.1
.2
.3
1/4
.4
3/8
.5
1/2
.6
ID

PT.	A-DIA.	H-II
1	1.0323	.7994
2	1.0300	.8016
3	1.0277	.8038
4	1.0254	.8060
5	1.0231	.8082
6	1.0207	.8104
7	1.0183	.8126
8	1.0159	.8148
9	1.0135	.8170
10	1.0111	.8192
11	1.0087	.8214
12	1.0063	.8235
13	1.0039	.8256
14	1.0015	.8277
15	.9991	.8298
16	.9967	.8319
17	.9943	.8339
18	.9918	.8360
19	.9893	.8381
20	.9868	.8402
21	.9843	.8423
22	.9817	.8444
23	.9791	.8466
24	.9765	.8487
25	.9739	.8508
26	.9712	.8529
27	.9685	.8550
28	.9658	.8572
29	.9631	.8593
30	.9603	.8614
31	.9575	.8636
32	.9547	.8657
33	.9518	.8679
34	.9489	.8701
35	.9460	.8722
36	.9431	.8744
37	.9401	.8766
38	.9371	.8787
39	.9341	.8809
40	.9310	.8831
41	.9279	.8853
42	.9248	.8875
43	.9217	.8896
44	.9185	.8918
45	.9153	.8940
46	.9121	.8962
47	.9089	.8984
48	.9056	.9006
49	.9023	.9028
50	.8990	.9050

IN. COUPL SUB ASM

PT.	A-DIA.	H-I
51	.8956	.6324
52	.8922	.6361
53	.8888	.6398
54	.8854	.6435
55	.8819	.6471
56	.8784	.6507
57	.8749	.6543
58	.8713	.6579
59	.8677	.6615
60	.8640	.6651
61	.8603	.6686
62	.8565	.6722
63	.8527	.6757
64	.8488	.6792
65	.8449	.6827
66	.8409	.6863
67	.8369	.6898
68	.8329	.6932
69	.8288	.6967
70	.8246	.7002
71	.8204	.7036
72	.8161	.7071
73	.8117	.7106
74	.8072	.7141
75	.8027	.7176
76	.7981	.7211
77	.7934	.7246
78	.7886	.7282
79	.7836	.7318
80	.7786	.7354
81	.7735	.7390
82	.7682	.7427
83	.7628	.7464
84	.7573	.7501
85	.7517	.7539

OUT PUT COUPL SUB ASM.



SEQUENCE OF OPERATIONS
1. CUT ROUGH BLANKS PER DWG. PF 750-194
2. HYDROGEN ANNEAL @1030°C FOR 40 MIN
3. FINISH MACHINE.

DIMENSIONS APPLY AT 69.7 ±0.5 °F

PF 750-150 R4

N/A PTS. 1 TO 7 AD 750-192
PTS 8 TO 79 AD 750-199
PTS 80 TO 85 AD 750-193

NOTE: SEE DWG (SEPIA) PF 750-150 R1 FOR GAGE DWG

DO NOT SCALE DRAWING

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES.
TOLERANCES BREAK EDGES .005-.015
FRACT. ±1/64 INT. COR .015 R. MAX.
DEC. ± .005
ANGLES ± 1/2° ALL SURFS. 16
SCALE 1:1

PT.	QUAN		MATERIAL
78-85	1 EA	PF 750-194	PT. 9
72-77			PT. 8
62-71			PT. 7
53-61			PT. 6
45-52			PT. 5
32-44			PT. 4
19-31			PT. 3
6-18			PT. 2
1-5			PT. 1

Exhibit 4: Disks for Accelerator Section.

4		ANNEAL @ 1030°C WAS 960°C	R.B.
3	R.L.	ALL PTS. WERE MADE FROM PF 750-011.	R.B.
2		ADDED BALL HEIGHTS	
1		DWG. UP WAS SK 11°, ADDED CHAMFER	

STANFORD LINEAR ACCELERATOR CENTER
U.S. ATOMIC ENERGY COMMISSION
STANFORD UNIVERSITY STANFORD, CALIFORNIA

PROPRIETARY DATA OF STANFORD UNIVERSITY AND/OR U. S. ATOMIC ENERGY COMMISSION. RECIPIENT SHALL NOT PUBLISH THE WITHIN INFORMATION WITHOUT SPECIFIC PERMISSION OF STANFORD UNIVERSITY.

DSGN. R. BORGHI
DFTR. R. LAURIE 5-31-63
APPROVALS

DISC - FINISH MACH. -M-

PF 750 - 150 R4

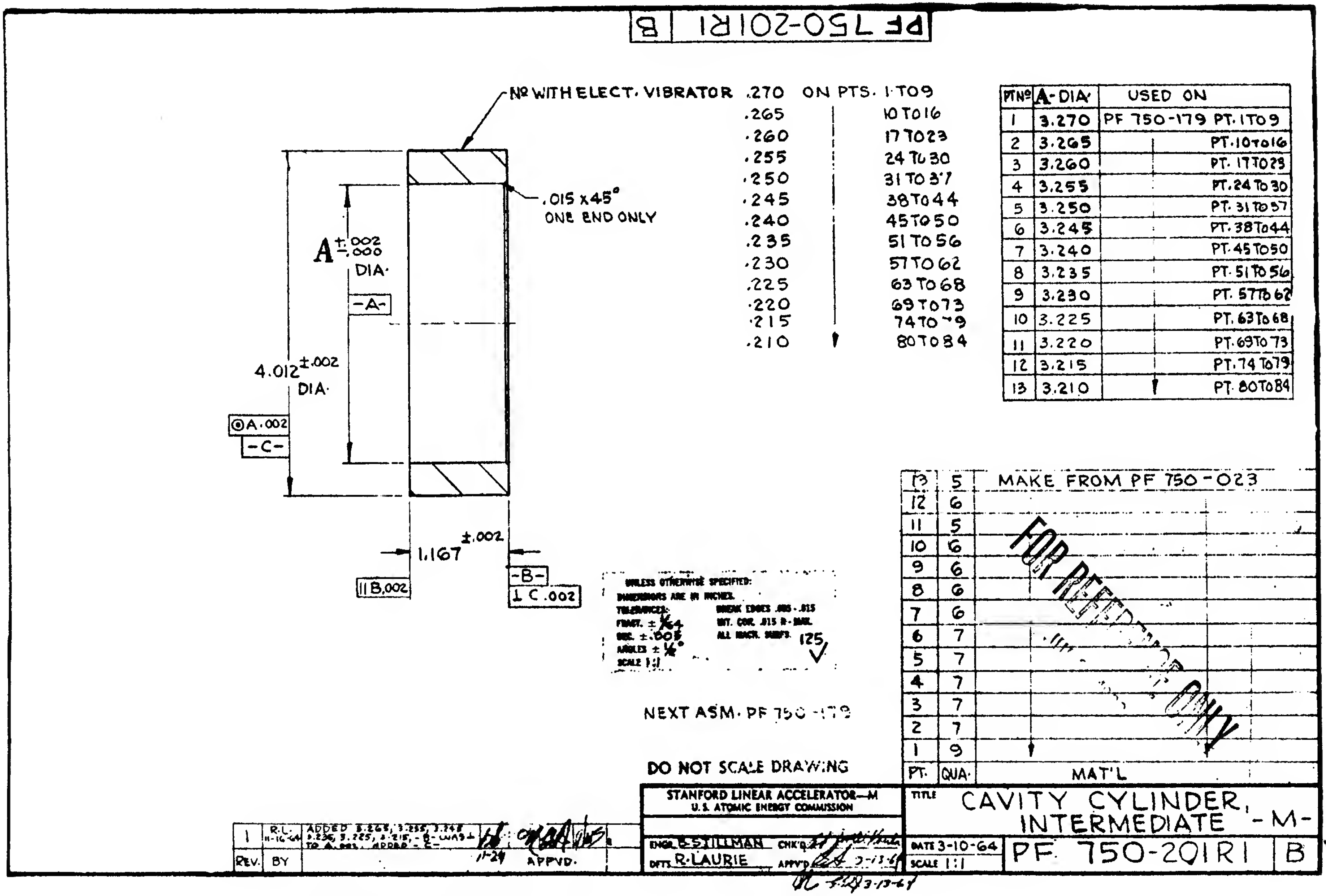


Exhibit 5: Cylinders for Accelerator Section.

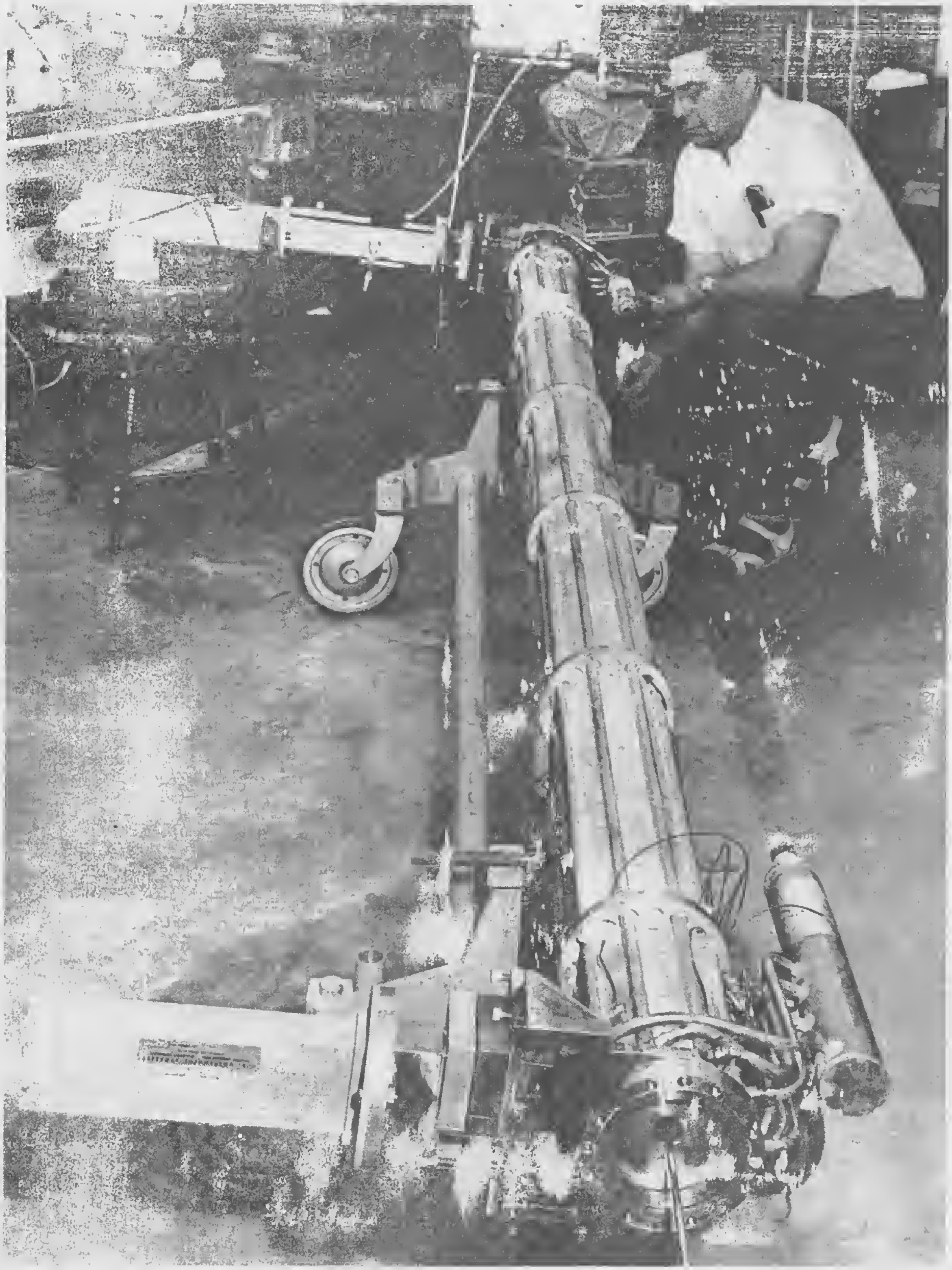


Exhibit 6: Eight Tube Constant Structure Test Section with a Single Manifold at Each End.

STANFORD LINEAR ACCELERATOR CENTER II (D)

Beam Pipe Cooling System

By the end of 1962 Al was making analytical studies of constant gradient accelerators with both 8 and 16 counterflow tubes. Samples of some of his calculations appear in Exhibit 1. These were made in December 1962 for a pipe with eight tubes of .34 inch i.d. and a water flow of 13 gpm. Earlier in the year he had made a broader parametric study of the uniform structure alternative, considering tube i.d.'s of .2, .3, .4, .5, and .6 inch. On the basis of these results he narrowed his studies for the constant gradient accelerator down to four commercially available tube i.d.'s -- .34, .36, .40 and .43 inch.

On the first page of Exhibit 1 he finds a value for the film conductance h from an empirical formula for fully established turbulent flow. At the top of this page he finds the water temperature rise over the ten foot length to be 7.1°F. On page 2 he finds the difference in temperature between the base of the tube and the water. Continuing, the water temperature difference between two adjacent tubes is determined at both ends, the middle, and the 1/4 and 3/4 points on a section, taking account of the fact that, since they are at different temperatures, heat flows from one tube to the other and thus less than 1/8th of the total heat generated flows to the hotter tube while more than 1/8th flows to the colder. In further calculations, not shown, he determines the axial temperature variation along the accelerator section.

Al determined temperature distributions within various sections of the accelerator pipe wall by using electrical analog models made of carbon impregnated Teledeltos conducting paper. To do this he needed the film conductance and water temperatures along the length of a tube from his calculations. Al said, "At this time we didn't have a computer, so an analog was the only practical way to get the temperature distributions inside the wall."

He cut a piece of the Teledeltos paper to the shape of a segment of the accelerator between the centerlines of two tubes and scaled to represent the conductivity of the OFHC copper as shown in Exhibits 2 and 3. Since the wires are attached to the paper only at a finite number of points, the paper is extended inside the pipe and tube walls and this portion is slotted so that current entering or leaving can flow only radially. The resistors in the leads from the tubes simulate the film conductance h. The voltage applied to the two cooling tubes can be varied to represent different water temperatures. Voltages at various points in the interior of the "wall", which are proportional to the temperatures, are read with a probe.

Al also used these conducting paper models to investigate changes in contact area between the tubes and the pipe; while the model in Exhibit 2 has a large radius at the junction representing a large brazed fillet, that in Exhibit 3 has been cut down to simulate a much smaller fillet. Various tube wall thicknesses were also checked with the analog. Typical results of measurements made on a conducting paper model representing a section taken at the inlet of one cooling tube and the outlet of the adjacent tube for a power input level of 13.1 kw is shown in Exhibit 4.

Before Al began working with the flux plot models he was worried that the value of h might vary around a tube because of the change in local temperature. However, he found from the models that peripheral variations in the value of h did not lead to significant axial temperature differences, so he did not have to try to determine if h actually did vary.

An amplification of the analytic and flux plot studies and some comments on the alternatives Al was then considering are contained in the memo reproduced in Exhibit 5.

Al now planned a more detailed comparison of the alternative tube configurations available to him, including those mentioned in the memo of Exhibit 5 and a one-piece tube and spacer consisting of a pedestal of varying cross-sectional area with a water passage formed integrally above it. He also considered variations on these ideas, for instance, squashing an initially round constant diameter tube to vary the velocity, a method analogous to tapering the tubes.

Al was still worried about how to compensate for entrance effects, as the curves in Exhibit 5 show, and about how to control the temperature in the 960 pipe sections. For this he thought he would need both a means of varying the water inlet temperature and a knowledge of the heat transfer characteristics of the jacket. He said, "At first we were thinking of sensing a temperature somewhere on each section and automatically regulating the water inlet temperature to that section as a function of this. But it would have meant almost a thousand control systems -- a lot of money. A little later we thought that since at Stage 1 a single klystron would feed four sections, all four could be controlled together. This got us down to only 250 control systems. But as the money got tighter we began to think about using one water system for each sector. A sector consists of 32 sections and is another natural division in the accelerator length because of the way the klystron power supplies are hooked up."

A Technical Note* written in 1962 by R. B. Neal, Associate Director of SLAC for the Technical Division, contained the following paragraphs on temperature control:

DISCUSSION OF SECTOR CONTROL

In the case of sector control of cooling water temperature, the temperature deviations of individual sections are the sum of three separate components:

(1) The error in the determination of the "correct" temperature at which the sector should be maintained;
(2) The error in the temperature of the water to the entire sector;
(3) The error in the temperature of the metal structures of individual sections.

The 'correct' base temperature for the sector cooling water depends upon the average power level. In this discussion, it is presumed that the klystrons in the sector are all operated at the same nominal beam voltage, pulse length, and pulse repetition rate. It is desired to maintain the accelerator sections at a constant temperature at all power levels. To accomplish this, the input cooling water temperature must be lower than the temperature of the metal structure of the accelerator section by an amount proportional to the average rf power input to the section. Among the several ways by which the correct input water temperature may be determined are the following: (a) by monitoring the temperature of the metal of the accelerator structure using appropriate sensing devices and maintaining the mean metal temperature constant[1] by adjusting the input cooling water tower temperature (for improved accuracy, the <u>average</u> temperature of all the sections comprising the sector should be maintained constant; the average might be determined, for example, by connecting all the sensing devices attached to the individual accelerator sections of the sector in series) (b) by monitoring the average output power of the dc supply connected to the sector modulators and by multiplying this output by the known klystron efficiency factor; (c) by measuring the water temperatures in the supply and return manifolds of the accelerator section and by adjusting the input water temperature to keep this phase difference constant. Of the four methods outlined here, method (d) should be most accurate and has the advantage that the temperature drop across the accelerator wall is automatically taken care of; however, it would probably be more expensive to instrument than the others. Method (a) would probably give sufficient accuracy with reasonable cost. The accuracy of methods (b) and (c) is affected by beam loading variations, but they might still be adequate.

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[1] It is important to take into account the temperature drop across the metal wall of the accelerator structure. For copper walls and for the parameters of the M Accelerator this temperature drop is given by $\delta T = 0.053\ t\ P_{KW}$, where t is the wall thickness in inches and P_{KW} is the average rf power input in kilowatts to the accelerator section. At the maximum Stage II power level of 21.6 kw and for t = 3/8 inch, T = 0.43°C. The pertinent temperature drop governing cavity dimensions is approximately T/2 which has a maximum Stage II value of 0.22°C (0.39°F).

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*DISCUSSION OF ACCELERATOR COOLING WATER REQUIREMENTS, TN 62-25, R. B. Neal, April 25, 1962

The magnitude or errors in class (2) depend upon the accuracy to which temperatures can be maintained in economically feasible control systems. Rogers in TN 62-20 states that a control system has been developed which can easily maintain water temperatures within $\pm$ 0.2°F. For the purpose of this note, it will be assumed that the sum of errors in classes (1) and (2) can be held to $\pm$ 0.4°F, except during transient periods.

Errors in class (3) arise primarily from variations in the average output power of individual klystrons. Since the difference in the temperature of the metal accelerator structure and the input cooling water temperature is proportional to the average rf power input, a deviation of input power from the standard value on which the input water temperature is based will result in a deviation in the temperature of the metal wall from the correct value. At the maximum Stage II power level, approximately 15 kw of rf power is dissipated in each accelerator section (in the absence of beam loading).

Lisin[1] has calculated and experimentally verified the temperature difference between the input cooling water and the metal accelerator structure at various values of power dissipation and water flow rates for a 10-foot accelerator structure equpped with 16 - 0.43 inch inside diameter pipes carrying cooling water in a counter-flow arrangement. He finds, for example, that for a power dissipation of 15 kw and a flow rate of 20 gallons/minute the temperature difference is proportional to power dissipation. However, for a given power dissipation, the temperature difference does not increase proportionally when the flow rate is decreased (e.g., at 15 kw dissipation and 10 gallons/minute flow the difference is 16.2°F). From the above results it is clear that, at a flow rate of 20 gallons/minute, variations of $\pm$ 10% in the output of individual klystrons (due to manufacturing tolerances, aging, variation in rf drive, etc.) from the nominal value would (at maximum Stage II power level) cause $\pm$ 1°F variations in the temperature of the accelerator sections receiving the power from the affected klystrons. The corresponding temperature variations in the case of 10 gallons/minute flow rate would be $\pm$ 1.6°F, etc.

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[1] A. V. Lisin, "Tentative Water Jacket Requirements", TN 62-23, Supplement No. 1, April 19, 1962.

Exhibit 1: Some Calculations Made by Al Lisin in December 1962. ECL 61-D Exhibit 1 Page 1

AVL 12/7/62

Convection Heat Transfer in Tubes (8 tubes)

Heat input $q = 13.5 \text{ kw} \times 3413 = 46,000$ Btu/hr.

Water flow $w = 13 \text{ gpm} \times 500 = 6500$ lb/hr

Water inlet temp $\approx 100°F$

$$c_p = .997 \text{ Btu/lb°F}$$

$$\therefore \theta = \frac{46,000}{.997 \times 6500} = 7.1°F$$

Water properties @	100°F	107°F	
c_p	.997	.997	B/lb°F
k	.364	.3645	B/hr ft °F
μ	1.65	1.61	lb/hr ft
ρ	62.0	61.9	lb/ft^3
$Pr = \frac{c_p \mu}{k}$	4.52	4.40	

Flow per tube = 813 lb/hr.

Tube ID = .340" = .0283'

Flow area $= \frac{\pi}{4} \times \overline{.0283}^2 = 6.30 \times 10^{-4}\ ft^2$

$G = w/A = 813/6.30 \times 10^{-4} = 1.291 \times 10^6\ lb/hr\text{-}ft^2$

$V = \frac{w}{A\rho} = \frac{1.291 \times 10^6}{3600 \times 62.0} = 5.78$ fps

$$Re = \frac{GD}{\mu} = \frac{1.291 \times 10^6 \times .0283}{1.65} = 22,100$$

For fully established turbulent flow of fluids with $1 < Pr < 20$

$$St\, Pr^{0.55} = .023\, Re^{-0.2}$$

$$\therefore \text{our } St = \frac{.023}{\underbrace{(4.46)^{.55}}_{2.27}\ \underbrace{(22,100)^{.2}}_{7.40}} = 1.37 \times 10^{-3}$$

$\frac{.02}{.015}$

$$St = \frac{h}{G c_p}$$

$$\therefore h = 1.37 \times 10^{-3} \times 1.291 \times 10^6 \times 0.997 = 1762\ B/hr\,ft^2\,°F$$

 AVL[2] 12/7/62

Fin Effect

Examining the geometry, we see that each tube is essentially one fin, 0.160" thick and 0.65" long.
The copper conductivity is 220 B/hr/ft °F
We found h to be 1762 B/hr/ft² °F

The fin efficiency is $f(ml)$

$$m = \sqrt{\frac{2h}{k\delta}} = \sqrt{\frac{2 \times 1762 \times 12}{220 \times 0.160}} = \sqrt{1202} = 34.7$$

$$ml = 34.7 \times \frac{0.65}{12} = 1.878 \qquad \therefore \eta = 51\%$$

(from Kays & London)

$\therefore$ at the tube inside surface:

$$q = hA\eta\Delta t$$

$$A_h \text{ per tube} = \pi D L = \pi \times \frac{.340}{12} \times 10 = 1.069 \text{ ft}^2$$

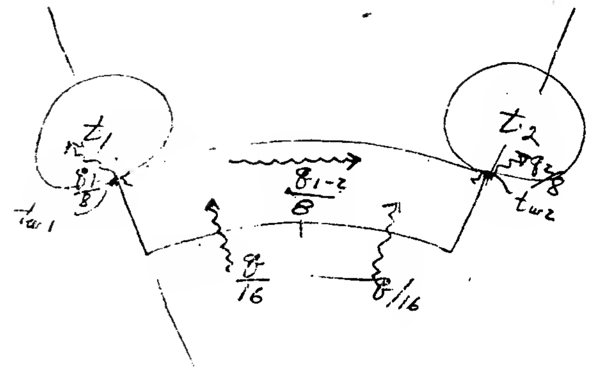
$\therefore$ for each tube (fin base to water Δt)

$$\Delta t = \frac{q\text{-tube}}{1762 \times 1.069 \times .51} = \frac{q/\text{tube}}{960} = \frac{q}{3840}$$

At the accelerator mid-point

$$\Delta t = \frac{13.5 \times 3413}{8 \times 960} = 6.0 \text{ °F}$$

Accelerator Wall



We can determine the heat transfer rate to each tube and the wall temp under each tube by superposition. Because of symmetry we can slice thru each tube. Then 1/16 of heat generated goes to each tube half. Since $t_1 \neq t_2$ there will also be heat transferred from one tube to the next. This heat transfer rate is $q_{1-2}/8$.

$$q_1 = \frac{q}{2} - q_{1-2}$$

$$\text{\&} \quad q_2 = \frac{q}{2} + q_{1-2}$$

now, $$q_{1-2} = 8\,\frac{kA}{x}(t_{w1} - t_{w2}) = 8\,\frac{t_{w1} - t_{w2}}{R_k}$$

$$\therefore \quad q_1 = \frac{q}{2} - 8\,\frac{t_{w1} - t_{w2}}{R_k}$$

$$\text{\&} \quad q_2 = \frac{q}{2} + 8\,\frac{t_{w1} - t_{w2}}{R_k}$$

also $$q_1 = 8hA\eta\,(t_{w1} - t_1) = \frac{8}{R_c}(t_{w1} - t_1)$$

$$t_{w1} = \frac{q_1 R_c}{8} + t_1$$

$$\text{\&} \quad q_2 = 8hA_2\eta\,(t_{w2} - t_2) = \frac{8}{R_c}(t_{w2} - t_2)$$

$$t_{w2} = \frac{q_2 R_c}{8} + t_2$$

$$q_1 = \frac{q}{2} - \frac{8}{R_k}\left[\frac{q_1 R_c}{8} + t_1 - \frac{q_2 R_c}{8} - t_2\right]$$

$$\left(1 + R_c\right) q_1 = \frac{q}{\;} + \frac{R_c}{\;}\, q_2 \quad - \frac{8}{\;}(t_1 - t_2)$$

and $$q_2 = \frac{q}{2} + \frac{S}{R_K}\left[\frac{q_1 R_c}{S} + t_1 - \frac{q_2 R_c}{S} - t_2\right]$$

$$\left(1+\frac{R_c}{R_K}\right) q_2 = \frac{q}{2} + \frac{R_c}{R_K} q_1 + \frac{S}{R_K}(t_1 - t_2)$$

Combining q_1 & q_2

$$\left(1+\frac{R_c}{R_K}\right) q_1 = \frac{q}{2} + \frac{R_c}{R_K\left(1+\frac{R_c}{R_K}\right)}\left[\frac{q}{2}+\frac{R_c}{R_K}q_1 + \frac{S}{R_K}(t_1-t_2)\right] - \frac{S}{R_K}(t_1-t_2)$$

$$\left(1+\frac{R_c}{R_K}\right)q_1 - \frac{R_c}{R_K} q_1 \frac{R_c}{R_K\left(1+\frac{R_c}{R_K}\right)} = \frac{q}{2}\left(1+\frac{R_c}{R_K\left(1+\frac{R_c}{R_K}\right)}\right) + \frac{S}{R_K}(t_1-t_2)\left(\frac{R_c}{R_K\left(1+\frac{R_c}{R_K}\right)}-1\right)$$

$$\frac{R_K+R_c}{R_K}q_1 - \frac{R_c}{R_K+R_c}\frac{R_c}{R_K}q_1 = \frac{q}{2}\left(1+\frac{R_c}{R_K+R_c}\right) + (t_1-t_2)\frac{S}{R_K}\left(\frac{R_c}{R_K+R_c}-1\right)$$

$$q_1 = \frac{q}{2} + \frac{S}{R_K}\frac{\left(\frac{R_c}{R_K+R_c}-1\right)}{\left(1+\frac{R_c}{R_K+R_c}\right)}(t_1-t_2)$$

& $$q_2 = q - q_1$$

Evaluating Constants:

$$R_c = \frac{1}{hA\eta} = \frac{1}{960} = .001022$$

R_k will have to be determined from examination of the geometry. Looking at the flow lines on the sketch, the flow width is essentially 1/4" & flow path length is 1 1/2"

$$R_k = \frac{x}{kA} = \frac{1.5}{220 \times 10 \times .25} = .00272$$

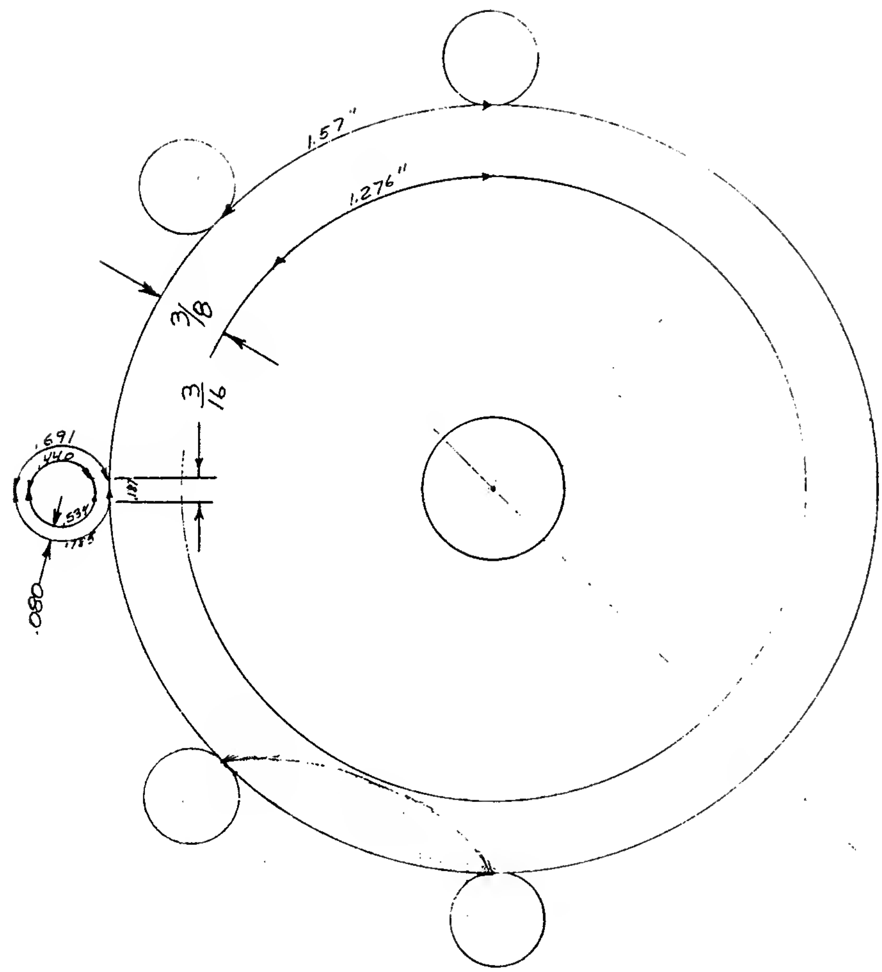
$$\frac{R_c}{R_c + R_k} = \frac{.001022}{.001022 + .00272} = .274$$

$$1.274\, q_1 = 1.274\, \frac{q}{2} + \frac{8}{.00272}\left(\underset{.726}{.274 - 1}\right)(t_1 - t_2)$$

2140

$$q_1 = \frac{q}{2} - 1680(t_1 - t_2)$$

$$q_2 = \frac{q}{2} + 1680(t_1 - t_2)$$

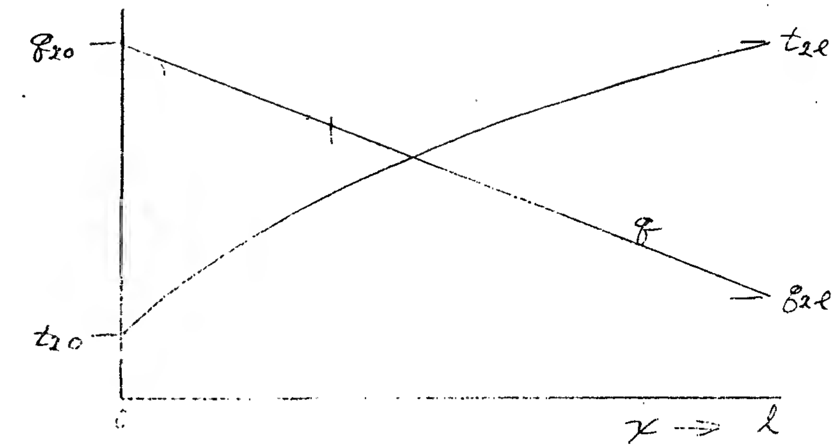

1.57"
1.276"
3/8
3/16
.691
.440
.534
.187
.080

Water Enthalpy Rise

For tube (2)

$$\frac{x}{\ell}\frac{q_{20}+q_2(x)}{2} = w_2 c_p \left(t_2(x) - t_{20}\right)$$

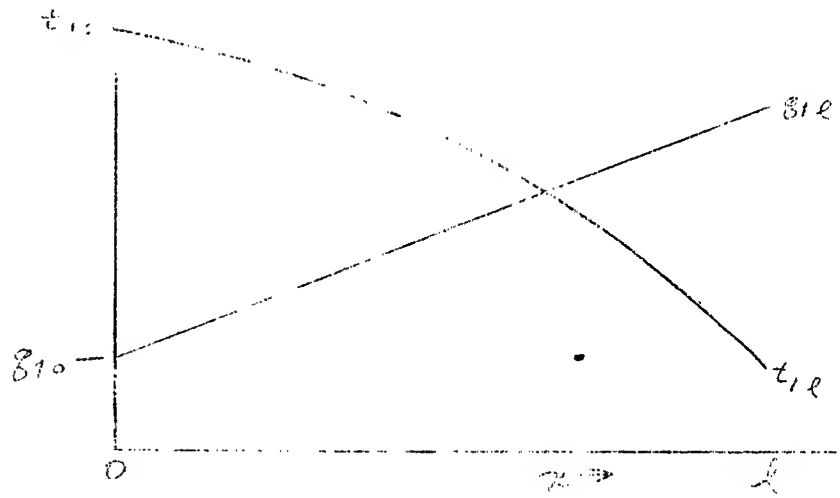
$$t_2(x) - t_{20} = \frac{q_{20}+q_2(x)}{2 w c_p}\frac{x}{\ell}$$



For tube (1)

$$\frac{x}{\ell}\frac{q_{10}+q_1(x)}{2} = w_1 c_p \left(t_{10} - t_1(x)\right)$$

$$t_{10} - t_1(x) = \frac{q_{10}+q_1(x)}{2 w c_p}\frac{x}{\ell}$$



Since

$$\frac{q_{10}+q_1(x)}{2} + \frac{q_{20}+q_2(x)}{2} = 2q$$

Combining yields

$$2q\frac{x}{\ell} = -w c_p \left(t_1(x) - t_2(x) + t_{20} - t_{10}\right)$$

Since $w_1 = w_2$; $t_{2\ell} - t_{10}$; $t_{1\ell} = t_{20}$,

$$2q\,x/\ell = -w c_p \left(t_1(x) - t_2(x) + \Theta\right)$$

or $t_2(x) - t_1(x) = 2\dfrac{q\,x/\ell}{w c_p} - \Theta$

$$t_2(x) - t_1(x) = \frac{2\times 13.5\times 3413}{13\times 500\times .997}\frac{x}{\ell} - 7.1 = 14.2\frac{x}{\ell} - 7.1$$

Calculation of conditions @ $x = 0, .25, .50$

@ $x = 0$:

$$t_{20} - t_{10} = -7.1\ °F$$

$$q_1 = q_{10} = \frac{46{,}000}{2} - 1680 \times 7.1 = 11{,}100$$

$$q_2 = q_{20} = 34{,}900$$

$$-t_1(x) + t_{10} = 0$$

$$-t_{20} + t_2(x) = 0$$

$$t_{w1} - t_{10} = \frac{11{,}100}{3840} = 2.88\ °F$$

$$t_{w2} - t_{20} = \frac{34{,}900}{3840} = 9.08\ °F$$

$$t_{w1} - t_{20} = 2.88 + 7.1 = 9.98\ °F$$

@ $x = .25$:

$$t_2 - t_1 = 14.2 \times .25 - 7.1 = 3.55 - 7.1 = -3.55$$

$$q_1 = 23{,}000 - 1680 \times 3.55 = 17{,}000$$

$$q_2 = 29{,}000$$

$$t_{10} - t_1(x) = \frac{11{,}100 + 17{,}000}{2 \times 3250 \times .997} \times .25 = 1.08\ °F$$

$$t_2(x) - t_{20} = \frac{34{,}900 + 29{,}000}{6500} \times .25 = 2.46\ °F$$

$$t_1(x) - t_{20} = 7.1 - 1.08 = 6.02\ °F$$

$$t_{w1}(x) - t_1(x) = \frac{17{,}000}{3840} = 4.43\ °F$$

$$t_{w1}(.25) - t_{20} = 10.45\ °F$$

$$t_{w2} - t_2 = \frac{29,000}{3840} = 7.56\,°F$$

$$t_{w2}^{(x=1)} - t_{20} = 7.36 + 2.46 = 10.02\,°F$$

@ $x = .50$:

$t_2 - t_1 = 0$

$\therefore \quad q_1 = q_2 = 23,000$

$$t_{10} - t_{1(x)} = \frac{11,100 + 23,000}{6500} \times .5 = 2.62\,°F$$

$$t_{2(x)} - t_{20} = \frac{34,900 + 23,000}{6500} \times .5 = 4.45\,°F$$

$$t_1(x) - t_{20} = 7.1 - 2.62 = 4.48\,°F$$

$$t_{w1}(x) - t_1(x) = \frac{23,000}{3840} = 6.00\,°F$$

$$t_{w1}(.5) - t_{20} = 10.48\,°F$$

$$t_{w2}(x) - t_2(x) = 6.00\,°F$$ % difference

$$t_{w2} - t_{20} = 10.45\,°F$$

@ $x = .75$: (for check of $x = .25$ step)

$t_2 - t_1 = 3.54$

$q_1 = 23000 + 6000 = 29,000$, $q_2 = 17,000$

$$t_{10} - t_1(x) = \frac{11,100 + 29,000}{6500} \times .75 = 4.62\,°F$$

$$t_1(x) - t_{20} = 7.1 - 4.62\,°F = 2.48\,°F$$

$$t_2(x) - t_{20} = \frac{34,900 + 17,000}{6500} \times .75 = 5.99\,°F$$

$t_{w1} - t(x) = \frac{29,000}{3840} = 7.56 \qquad t_{w1} - t_{20} = 10.04\,°F$

$t_{w2} - t_2(x) = \frac{17000}{3840} = 4.43\,°F \qquad t_{w2} - t_{20} = 10.42\,°F$

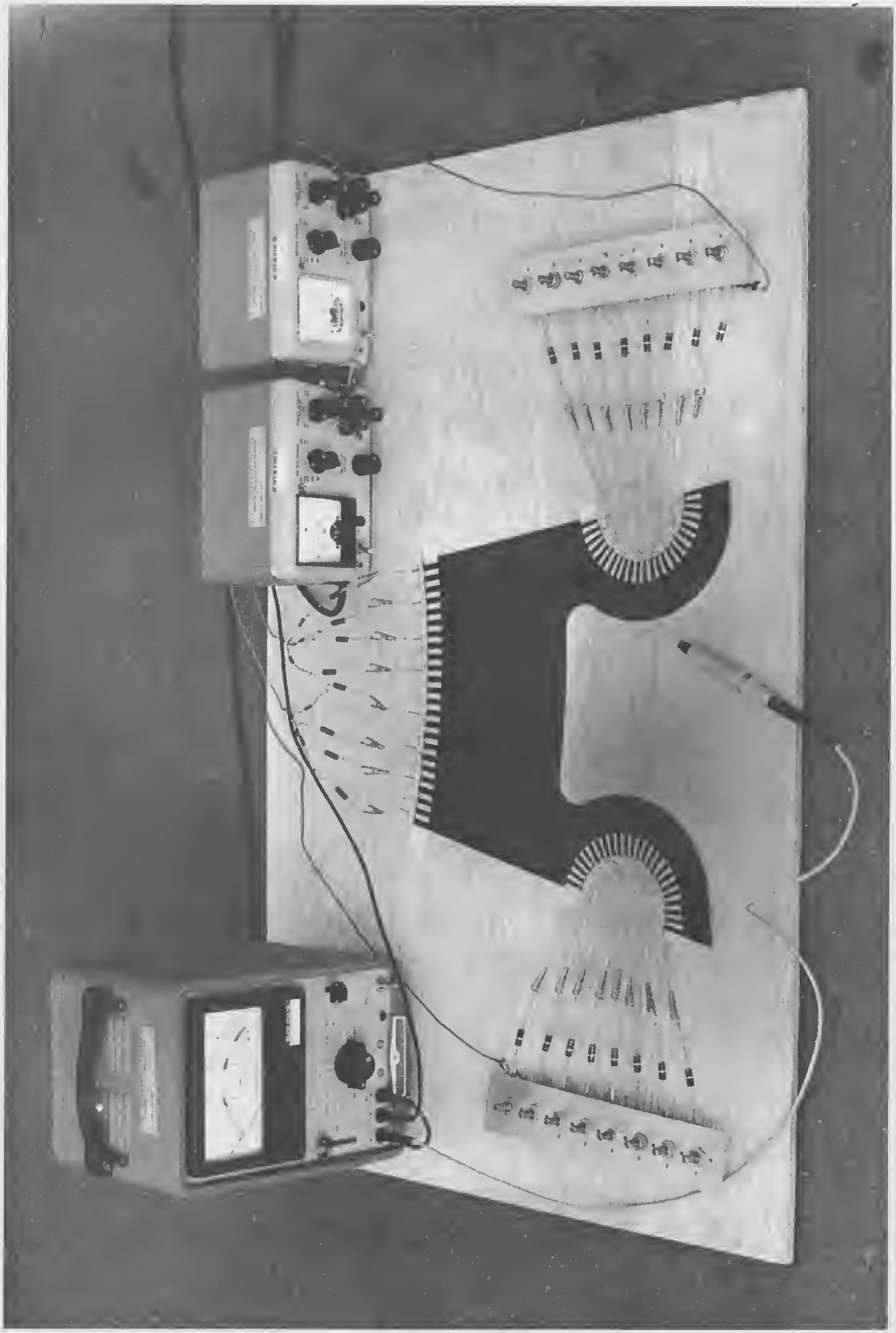


Exhibit 2: Conducting Paper Model for the 16 Tube Case.

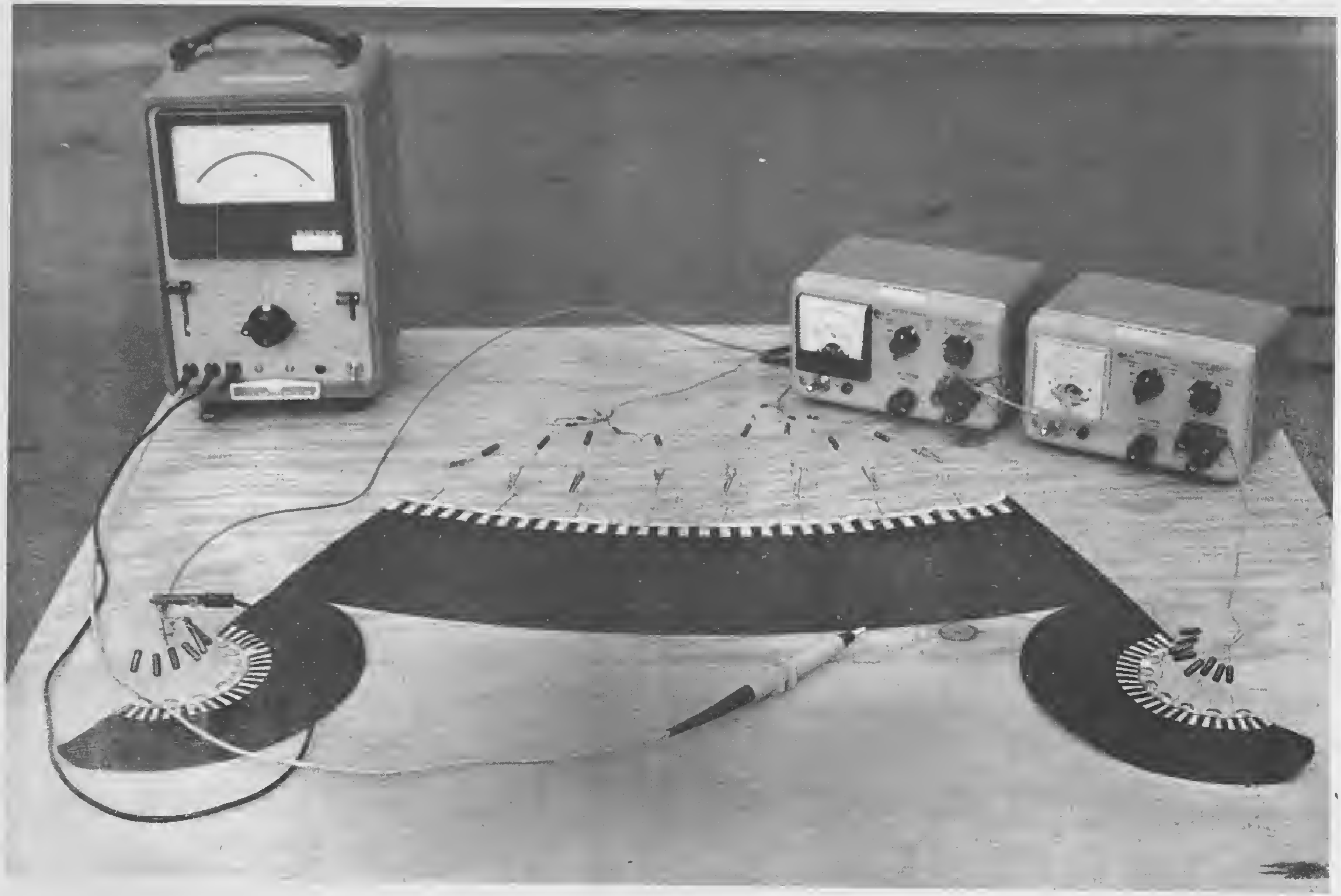


Exhibit 3: Conducting Paper Model for the 8 Tube Case.

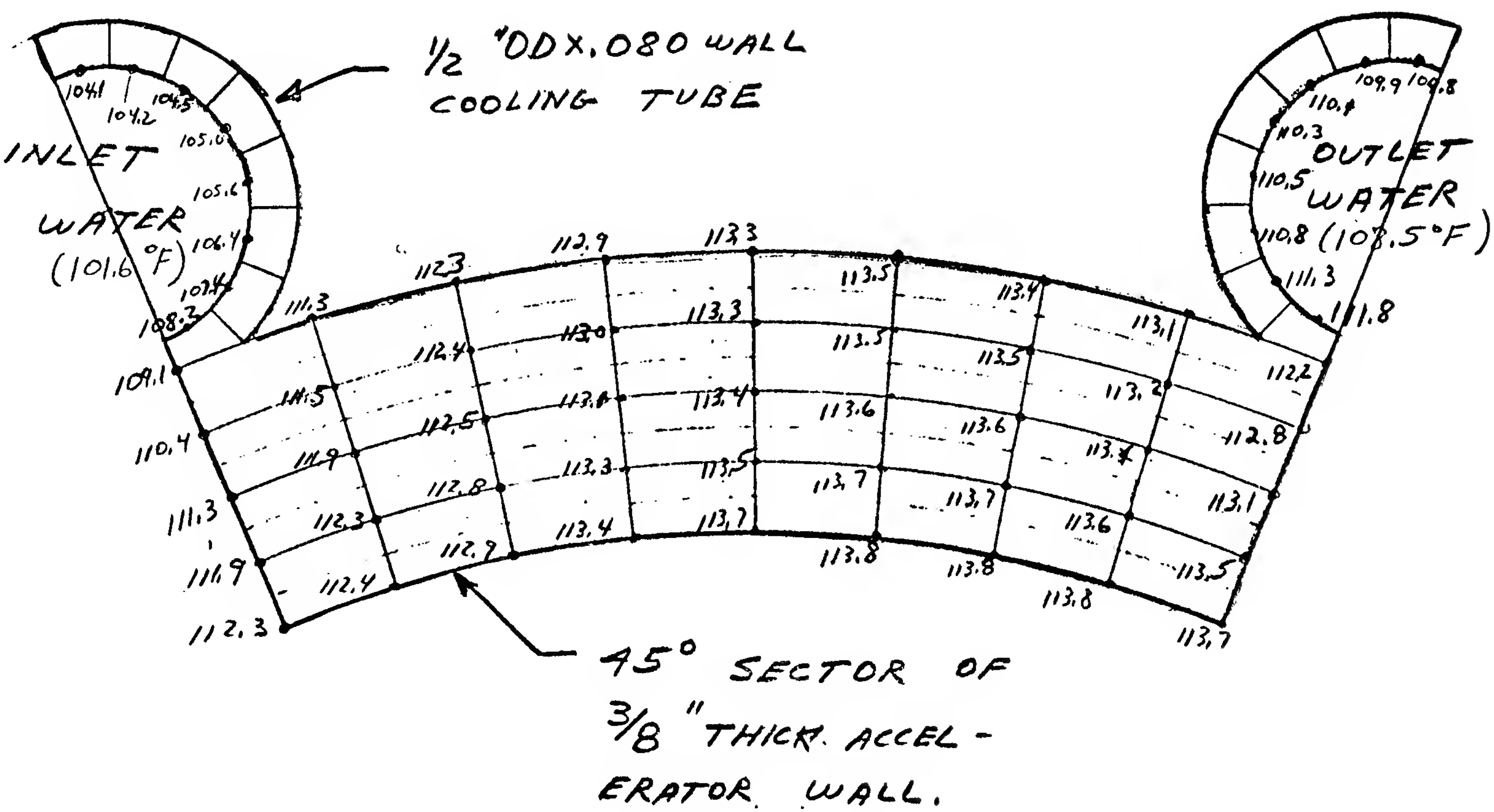


Exhibit 4: Flux Plot of Results from Conducting Paper Model for the 8 Tube Case. Section at Midpoint of 10 Foot Cooling Tube, Power Input 13.1 Kw.

25 January 1963

Arnold Eldredge

A. V. Lisin

Results of Recent Water Jacket Analyses

Since our meeting with Professor Kays, I have studied the counterflow tube type of water jacket further and have considered minor variations of the basic arrangement. Flux plots were made, using conducting paper models, to determine temperature distributions in the accelerator pipe and the volume average temperatures at the midpoint and ends of the pipe. Flux plots were made of several geometric configurations and at several water flow rates with constant diameter tubes. Entrance effects were not considered in the flux plots but were corrected for later. Data of Boelter[1], corrected to our conditions using data given me by Kays[2], was used to account for entrance effects. The effect of heat flux asymmetry on the convection conductance was considered in the flux plots. The magnitude of the correction was determined from Reynolds' analysis[3]. The correction was found to have a negligible effect on the temperature distribution.

The results of the flux plots are believed to be quite accurate. With the large scale model and the 1% voltmeter used, better than 0.1°F accuracy was achieved. For eight constant diameter tubes with a .080 inch wall and a flow of 13 gpm, the volume average temperature is about 2.5°F higher at the midpoint than at the pipe ends. The largest temperature variation occurs because of the hydrodynamic and thermal entrance effects. The average temperatures of the cavities which are unaffected by the entrance effects are all within 0.8°F.

Since heat is being removed from the accelerator section essentially along 8 lines, peripheral as well as radial temperature gradients exist. At the midpoint of the accelerator, the highest temperature, at the inside surface, is .81°F higher than the average and the minimum temperature, just under a cooling tube, is 2.01°F below the average. The maximum to average and average to minimum temperature differences are 1.11°F and 3.74°F respectively near the accelerator ends.

The expected average wall temperature distribution is shown in Figure 1. The band shows the temperature limits expected for the following tolerances:

Tube Wall Thickness	.080 ± .0035"	±.08°F
Tube ID	.340 ± .005"	±.07°F
Braze Width	.250 ± .032"	±.23°F
Braze Height	.002 ± .002"	±.04°F
Water Flow	13 ± .26 gpm	±.13°F
Total		±.55°F

The total tolerance buildup assumes, of course, that all tolerances are at their extremes at the same time. Heat conduction in the longitudinal direction has been neglected and this will help to decrease the tolerance effects. The uncertainty in trying to predict the convective heat transfer conductance in the entry region of the cooling tube is quite high. In addition, the heat transfer characteristics in the entry region are not as stable as they are after thermal and hydrodynamic boundary layers have become established. Therefore, the temperature uncertainty might widen at the accelerator ends by about a factor of 2.

The flux plotting results for the above conditions and for the other conditions tested are shown in Table I. The average temperatures ($\bar{t}$) are referred to the inlet water temperature (t_o). The last column shows the difference in average temperatures from the middle to the ends. Note that the case of 8 tubes with .160" wall and 1/4" contact is extrapolated from other measured results.

The table shows that increasing the wall thickness to .160" while keeping the cooling tube ID at .340" lowers the average temperatures by about .4°F but does not appreciably affect the middle to end temperature difference. Therefore, there would appear to be little advantage to increasing the cooling tube wall thickness. The use of 16 cooling tubes with a 13 gpm flow rate reduces the accelerator temperature level, but does not reduce the middle to end temperature difference. Therefore, there appears to be no advantage to increasing the number of tubes. In fact, the temperatures uncertainty would increase substantially because the flow per tube and therefore the Reynolds' number would be reduced to the point where the flow is not sufficiently far into the turbulent region to assure a stable convection conductance.

For 16 tubes and 26 gpm, the temperature level of the accelerator is only about 5-1/2°F above the inlet water temperature. The difference between middle and end average temperatures is about .44°F for a 1/4" wide braze. Here again, the braze joint width does not significantly effect the middle to end temperature difference although it does effect the temperature level of the pipe. Figure 2 shows the longitudinal variation in the volume average temperature for a 1/4" braze width. The temperature is seen to be much more uniform than for 13 gpm, 8 tube case. The cost of increasing the flow from 13 gpm to 26 gpm is $417,000, according to Hall[4].

In order to yield the same volume average temperature all along the accelerator, the cooling tube could be tapered so that the convection conductance increased along each cooling tube in a manner which would just correct for the water temperature rise. A tube taper which would yield a constant wall temperature at a flow rate of 13 gpm is shown in Figure 3. It might be pointed out that there is some uncertainty in the mind of Professor Kays about the behavior of the convection conductance in a flow passage which is continually decreasing in size. A further problem might be the fact that the tube size is so large at the inlet end that the flow is barely into the turbulent region and the convection conductance may not be stable.

A vendor was contacted to determine whether or not the desired taper could be put into our cooling tubes. By hydroforming, he felt that the tube ID could be increased to about 1/2 inch in a single operation if the tube were fully annealed. Further expansion of the tube would require another anneal. The cost of expanding the tube once would be about $7.50 each and dies would cost a couple thousand dollars. On the basis of this estimate it would cost about $16 per tube to obtain the desired taper. A concave surface to conform to the accelerator pipe curvature would be formed into the tube during the tapering process.

The possibility of placing the cooling tubes on fins, as mentioned by Professor Kays, was also considered. A tapered fin could be used to yield a uniform accelerator temperature. Such a tapered fin is shown in Figures 4 and 5. An estimate from a single vendor indicates that in large quantities such fins would cost less than $5 to machine on all four sides. The fin offers the advantage of making the variable heat transfer resistance in a conduction path. The thermal conductivity of a solid material is quite uniform, predictable, and repeatable and our ability to predict the heat transfer resistance of a simple conduction path is good. Thus, the uncertainties in the convection conductance associated with a tapered tube are removed. The disadvantage of the fin is the additional part which must be handled and the additional braze joint. Even with the added braze joint, the tapered fin with a straight tube will probably cost less installed than the tapered tube.

An alternate type of tapered fin might be one made in the shape of an "I" and on which the web thickness is tapered. The use of an "I" shaped fin would reduce temperature variation in the accelerator pipe due to variations in the braze joint width because the braze width (flange width) can be made substantially greater than the fin width. With a reduced heat flux across the braze joint, the effect of braze width on temperature will be decreased. A further advantage of the "I" fin might be the fact that all tubes are parallel so that the same support blocks can be used at any location along the accelerator pipe.

Assuming that either the tapered tube or tapered fin concept is adopted, the final taper dimensions in the tube entrance region will have to be verified by experiment. The tapered fin could be set up and varied quite readily by filing the fin until the desired temperature is attained. The tapered tube would be considerably more difficult to modify.

On the basis of the above, I would recommend that for greatest reliability and temperature uniformity, 8 straight tubes with .080 inch wall, brazed on an "I" shaped fin be adopted for the accelerator water jacket.

A. V. Lisin

AVL:mab

REFERENCES

1. Boelter, Young, Iverson. "Distribution of Heat Transfer Rate in the Entrance Section of a Circular Tube." NACA TN 1451

2. Kays. Heat Transfer Lecture Notes

3. Reynolds. "Turbulent Heat Transfer in a Circular Tube with Variable Circumferencial Heat Flux" (unpublished)

4. Hall. "Estimated Cost of Added Cooling Water for Accelerator Tubes" (memo to A. Eldredge, dated January 3, 1963)

TABLE I

SUMMARY OF FLUX PLOT RESULTS

No. Tubes	Flow	Tube Wall	Contact Width	$\bar{t} - t_o$ $x/\ell = .5$	$\bar{t} - t_o$ $x/\ell = 1$	$\bar{t}_{x/\ell = .5} - \bar{t}_{x/\ell = 1}$
8	13	.080"	1/4"	11.81	10.91	0.90°F
		.080"	3/16"		11.39	
		.160"	3/16"	11.90	11.01	0.89°F
	extrapolated	.160"	1/4"	11.42	10.53	0.89°F
16	13	.080"	3/16"	9.67	8.82	0.85°F
			.280"	9.29		
			.367"	9.02	8.06	0.96°F
16	26	.080"	3/16"	5.85	5.39	0.46°F
			.280"	5.53	5.10	0.43°F
			.367"	5.30	4.83	0.47°F

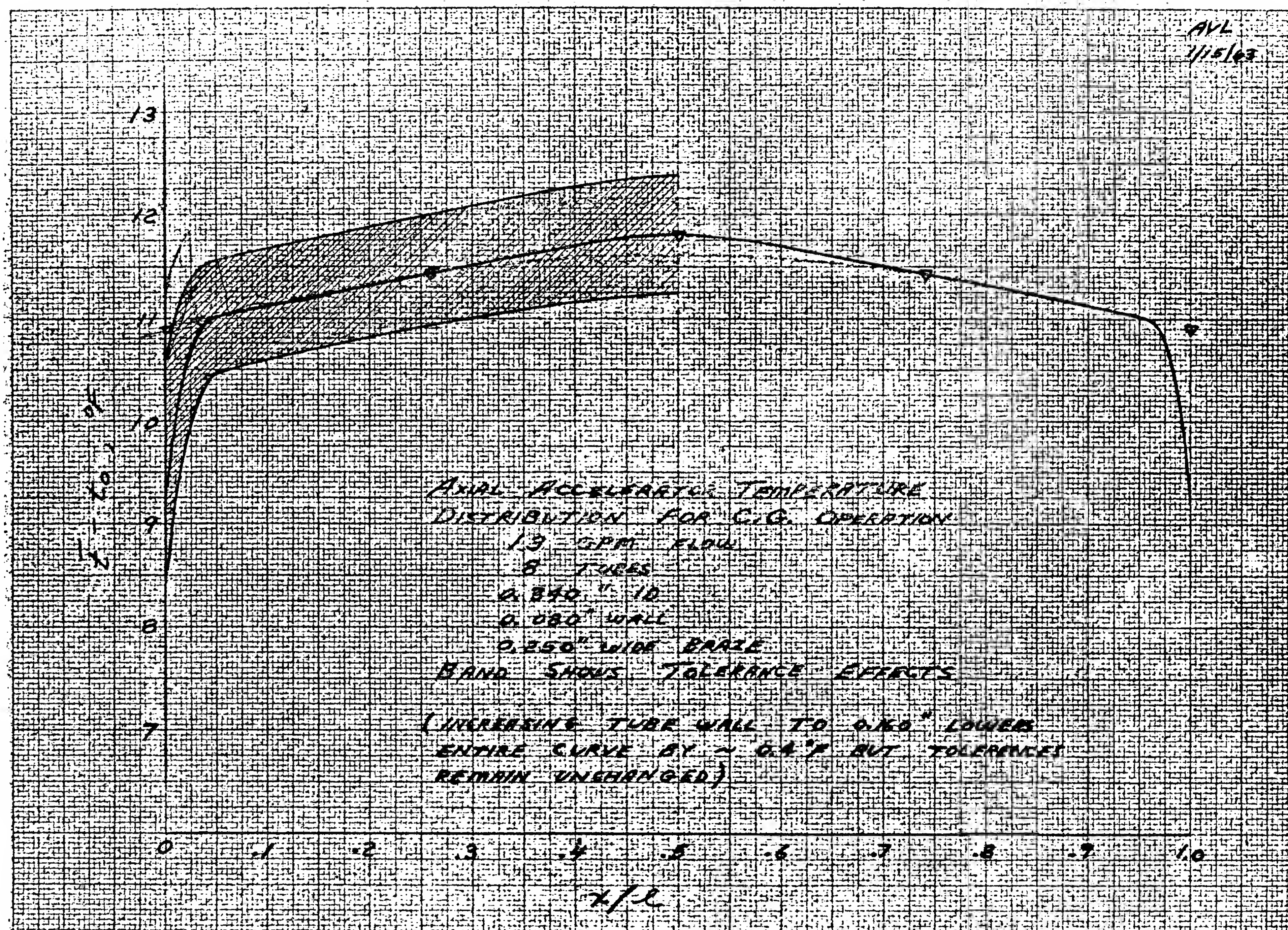

AVL
1/15/63
13
12
11
10
9
8
7
0
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0
x/l
AXIAL ACCELERATOR TEMPERATURE
DISTRIBUTION FOR C.G. OPERATION
1.3 GPM FLOW
8 TUBES
0.340" ID
0.080" WALL
0.250" WIDE BRAZE
BAND SHOWS TOLERANCE EFFECTS
(INCREASING TUBE WALL TO 0.160" LOWERS
ENTIRE CURVE BY ~ 0.4°F BUT TOLERANCES
REMAIN UNCHANGED)

FIG. 1

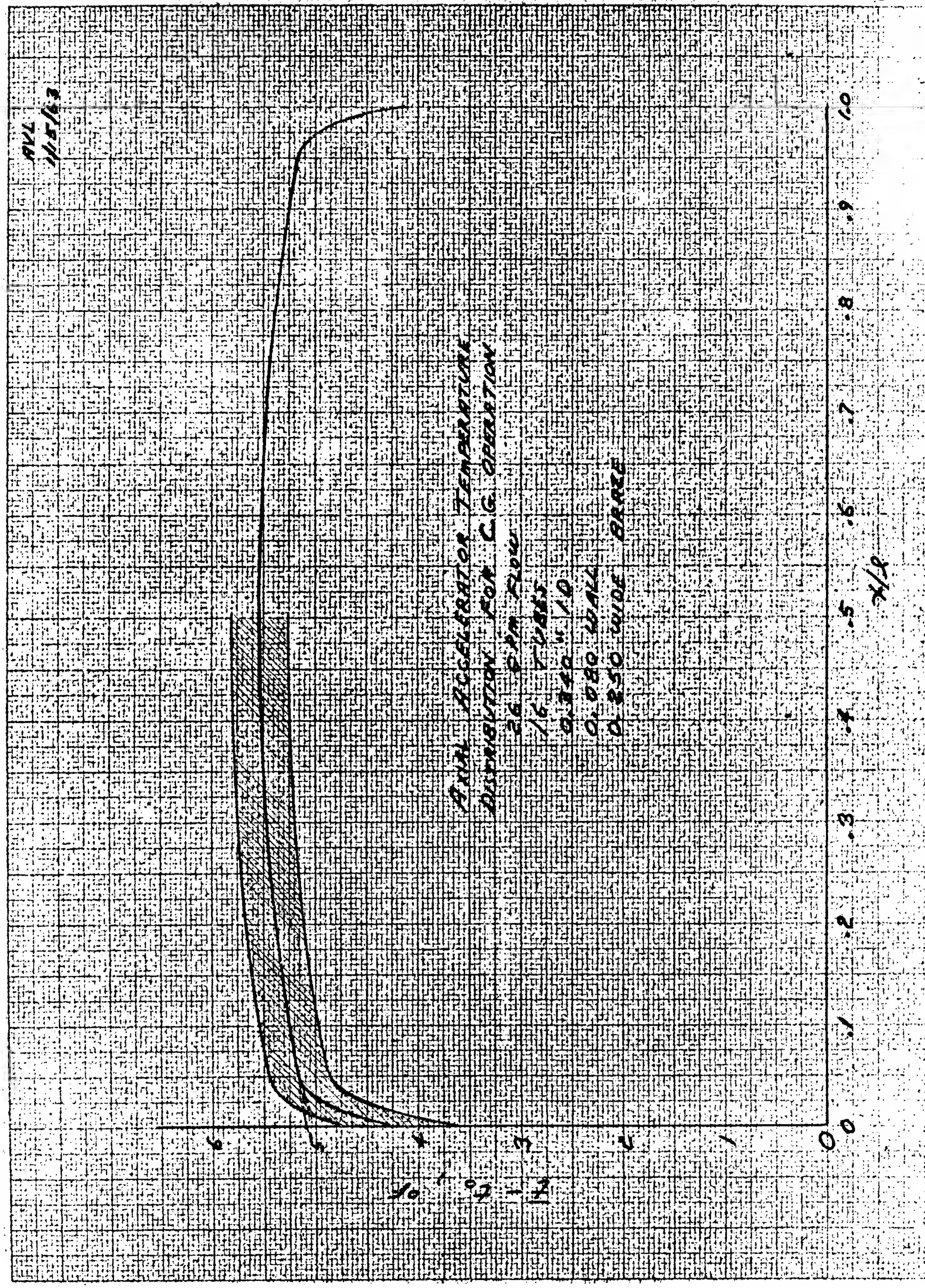

AVL
11/5/63
AXIAL ACCELERATOR TEMPERATURE
DISTRIBUTION FOR C.G. OPERATION
26 GPM FLOW
16 TUBES
0.240" I.D
0.080 WALL
0.250 WIDE BRAZE
0
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0
x/l
0
1
2
3
4
5
6

FIG. 2

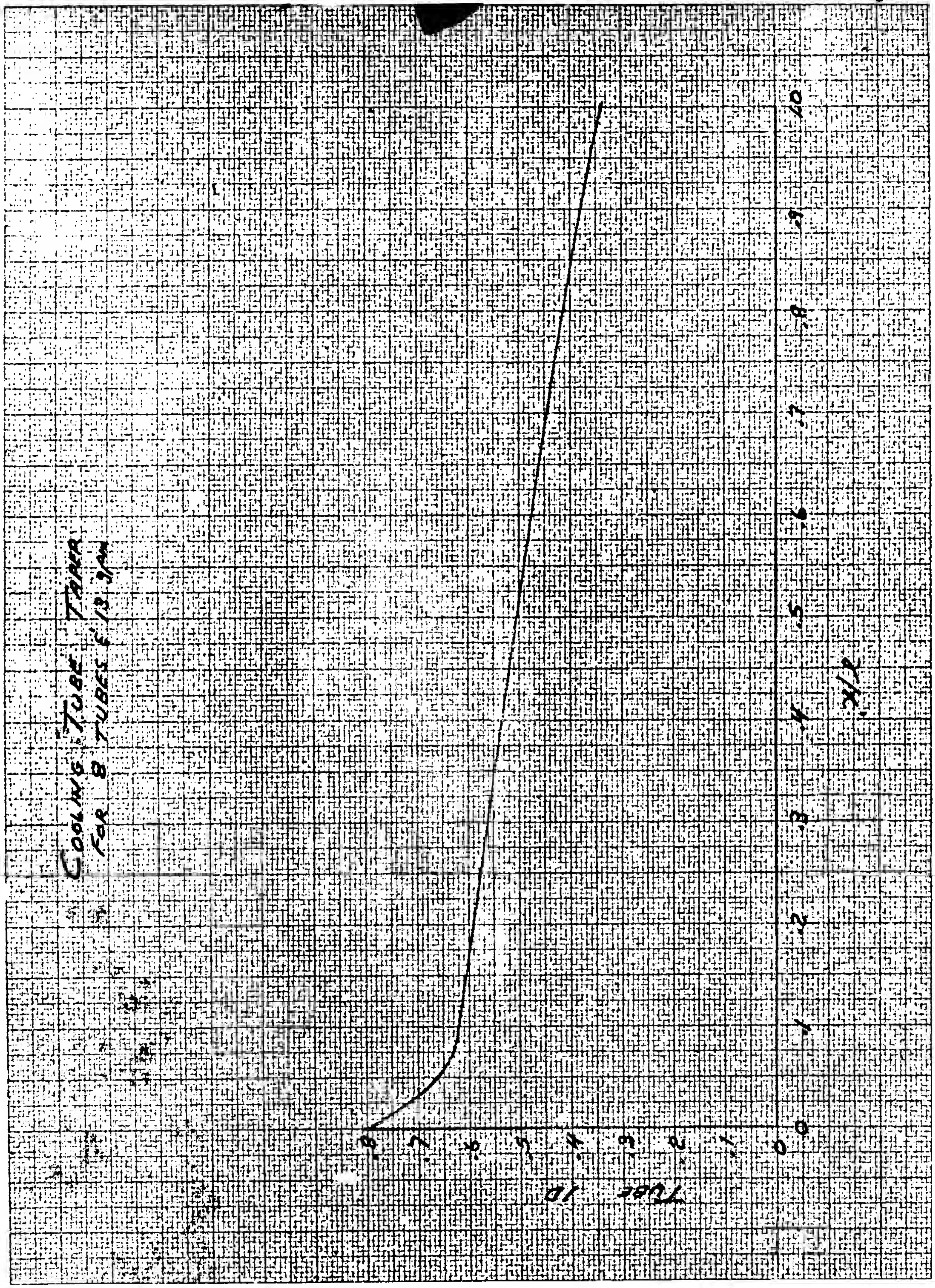
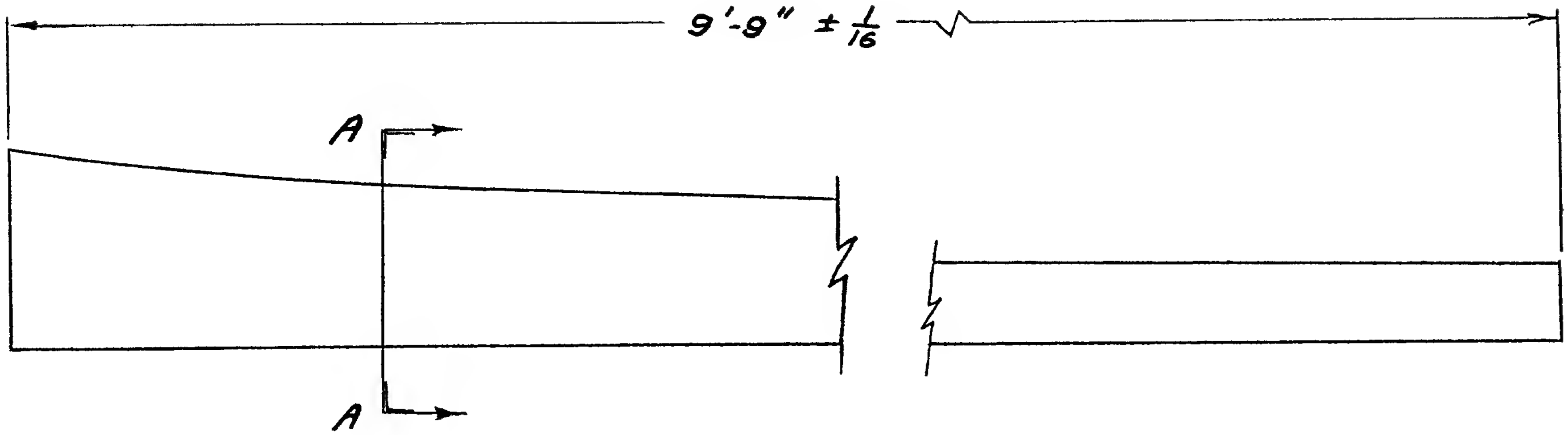
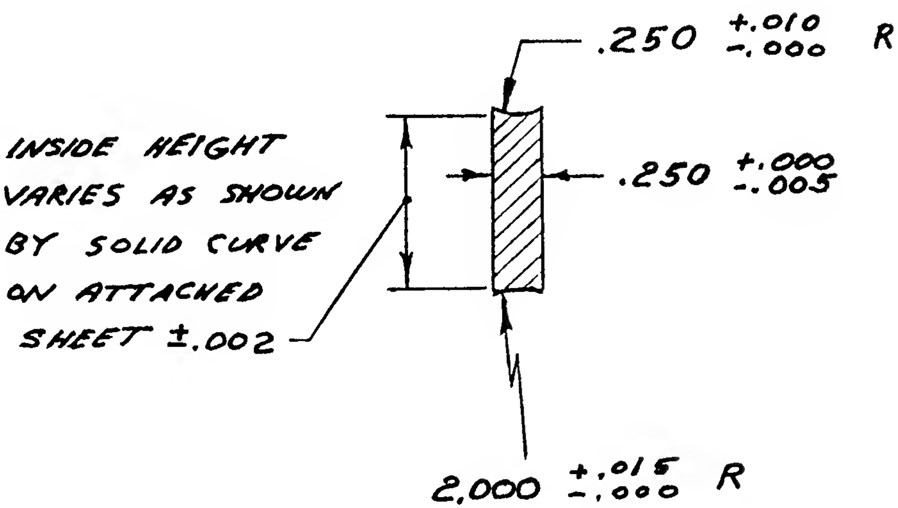


FIG 3





SECTION A-A

COOLING TUBE SPACER

FULL SIZE
MATERIAL - OFHC COPPER
A.V.L.
1/21/63

FIG. 4

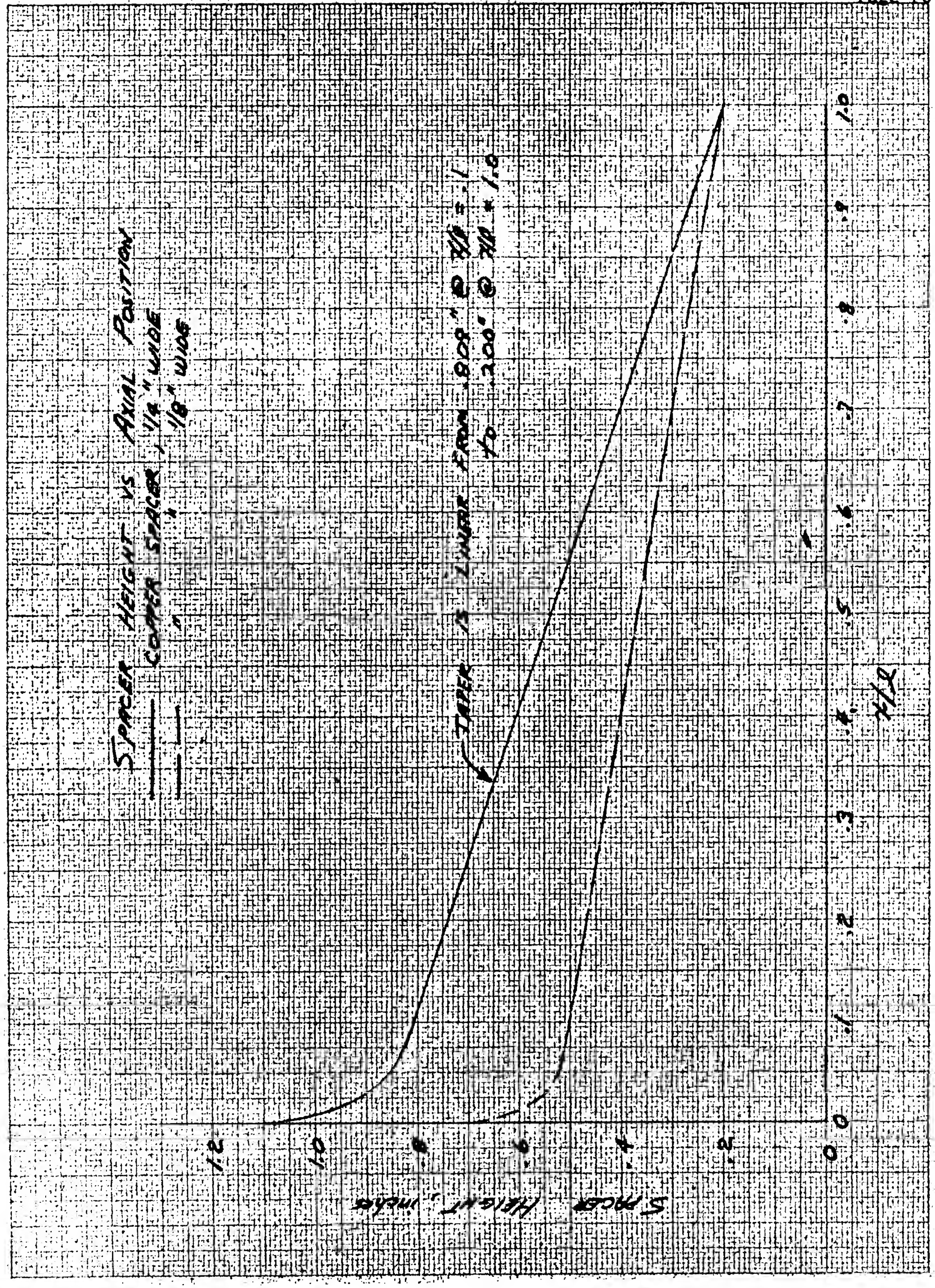

SPACER HEIGHT VS AXIAL POSITION
COPPER SPACER, 1/4" WIDE
" " 1/8" WIDE
TAPER IS LINEAR FROM .008" @ X/L = .1 TO .200" @ X/L = 1.0
X/L
SPACER HEIGHT, INCHES
0
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0
1.2
1.0
.8
.6
.4
.2
0

FIG 5

Tube Tapering

If we can make the tube base temp be constant along its entire length, then the temp profile within pipe will be [illegible] along its length & so the circumf. avg temp will be [illegible]. So we should taper tube to vary h in such a way to achieve above objective.

From the temp. plot for 8 straight tubes it [illegible] must raise temp at $x/\ell = 0$ by 1.3 °F [illegible] by 0.4°F.

To do this we must decrease $hA\eta$ for 75% [illegible] & ~90% at [illegible]

Let's try reducing $hA\eta$ until we get a constant wall temp, then determine what has happened to [illegible].

@ $x=0$

75% reduction in $hA\eta$ in tube 2

R_c becomes $\frac{.001022}{.75} = .001365$

$$\frac{R_c}{R_c + R_k} = \frac{.001365}{.001365 + .00272} = .334$$

$$\therefore q_2 = \frac{q}{2} - \frac{S}{.00272} \frac{(-.666)}{1.334} (t_1 - t_2)$$

$$= \frac{q}{2} + 1470 (t_1 - t_2)$$

$t_2 - t_1 = -7.1\ °F$

$q_1 = q_{10} = 23,000 + 10400 = 33,400$

$q_2 = q_{20} = 12,600$

$$t_{w_1} - t_{10} = \frac{12,600}{3840} = 3.29\,°F$$

$$t_{w_1} - t_{20} = 7.1 + 3.29 \quad 10.39\,°F$$

$$t_{w_2} - t_{20} = \frac{33,400}{.75 \times 3840} = 11.6\,°F$$

It appears that 75% is too low, so try 85%

$$R_c = \frac{.001022}{.85} = .001205$$

$$\frac{R_c}{R_c + R_K} = \frac{.001205}{.001205 + .00272} = 0.307$$

$$q_2 = \frac{q}{2} + \frac{8}{.00272}\,\frac{.693}{1.307}\,(t_1 - t_2) = \frac{q}{2} + 1562\,(t_1 - t_2)$$

$$q_1 = 23,000 - 11,100 = 11,900$$

$$q_2 = 34,100$$

$$t_{w_1} - t_{10} = \frac{11,900}{3840} = 3.10\,°F$$

$$t_{w_1} - t_{20} = 3.1 + 7.1 = 10.2\,°F$$

$$t_{w_2} - t_{20} = \frac{34100}{.85 \times 3840} = 10.45\,°F$$

@ $x/l = .25$ Let's try 95%

$$R_c = \frac{.001022}{.95} = .001077$$

$$\frac{R_c}{R_c + R_K} = \frac{.001077}{.001077 + .00272} = .284$$

$$q_2 = \frac{q}{2} + \frac{8}{.00272}\,\frac{.716}{1.284}\,(t_1 - t_2) = \frac{q}{2} + 1640\,(t_1 - t_2)$$

$$t_2 - t_1 \cong -3.55\ °F$$

$$q_2 = 23{,}000 - 5800 = 17{,}200$$

$$q_1 = 28{,}800$$

$$t_{w1} - t_1 = \frac{17{,}200}{3840} = 4.48\ °F$$

$$t_{w2} - t_2 = \frac{28{,}800}{.95 \times 3840} = 7.91\ °F$$

$$t_{w1} - t_{20} \approx 10.5\ °F$$

$$t_{w2} - t_{20} \approx 10.37\ °F$$

The calculation are only approximate as we assumed for the [illegible] determination that q varied linearly with X [illegible]. We are deviating from linearity by only a few % so [illegible] bad. ([illegible] °F).

Putting these new found temperatures [illegible] makes it [illegible]. Best, and is still low so try 83% [illegible]

$$R_c = \frac{.001022}{.83} = .001235$$

$$\frac{R_c}{R_k + R_c} = \frac{.001235}{.001235 + .00272} = .313$$

$$q_2 = \frac{q}{2} + \frac{8}{.00272}\,\frac{.687}{1.313}\,(t_1 - t_2) = \frac{q}{2} + 1540\,(t_1 - t_2)$$

$$q_2 = 23{,}000 - 10900 = 12{,}100$$

$$q_1 = 33{,}900$$

$$t_{w1} - t_{10} = \frac{12,100}{3840} = 3.16\,°F$$

$$t_{w2} - t_{20} = \frac{33,000}{.183 \times 3840} = 10.63\,°F$$

$$t_{w1} - t_{20} = 7.10 + 3.16 = 10.26$$

Determining Amount of Tape required

$$R_e = \frac{G D}{\mu} = 22,100 \, \frac{.340}{1D} = \frac{7510}{1D}$$

$$St = \frac{.023}{2.27 \times R_e^{.2}} = \frac{.0101}{R_e^{.2}}$$

$$G = 1.291 \times 10^6 \left(\frac{.340}{1D}\right)^2 = .1493 \times 10^6 \left(\frac{1}{1D}\right)^2$$

$$h = G\, c_p\, St = .997\, G\, St$$

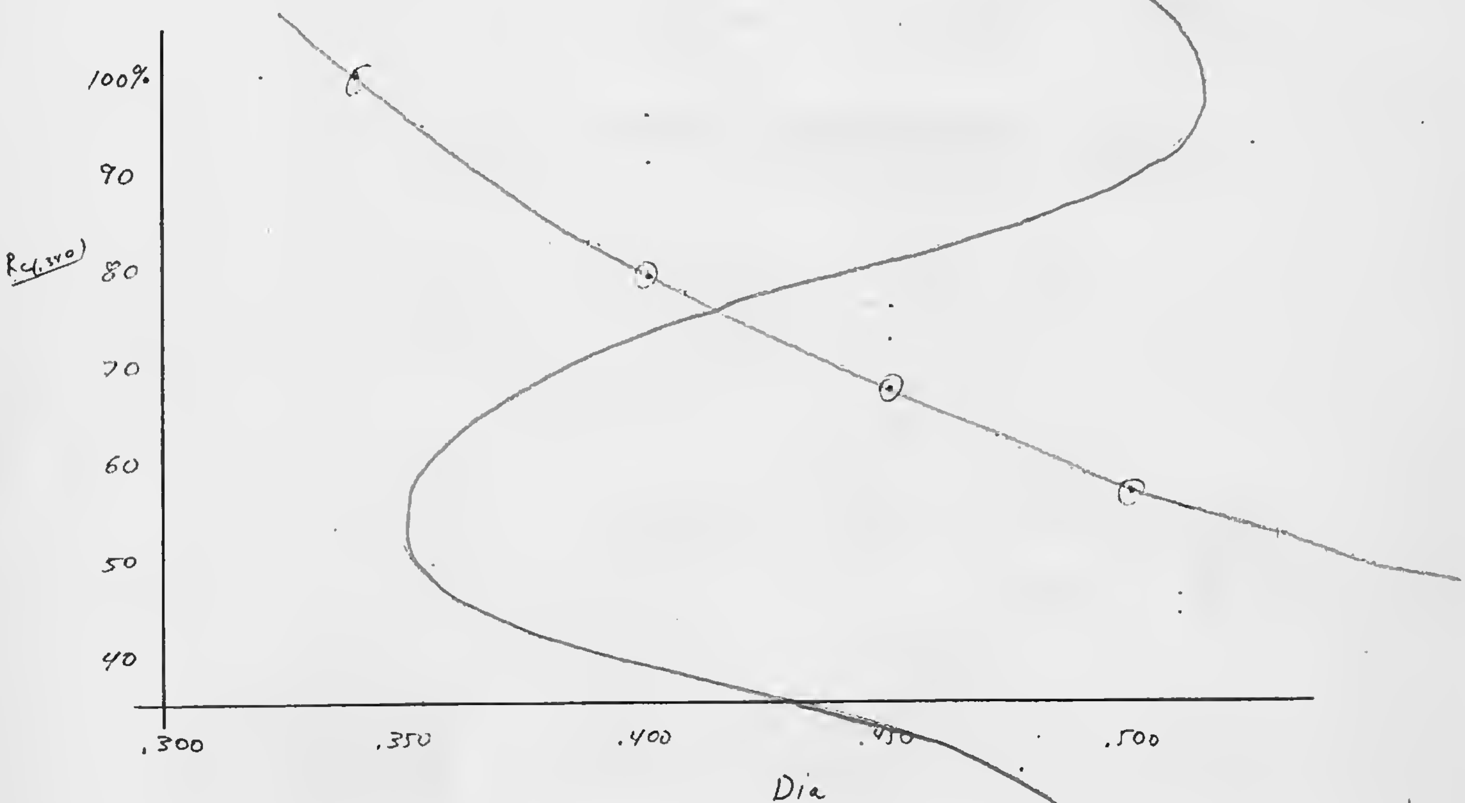
$$m = \sqrt{\frac{2h}{k\delta}} \qquad \delta = \frac{.340}{1D} \times .10$$

$$\ell = \frac{\pi \bar{D}}{2} = \frac{1D}{.340} \times .542 =$$

$$A = 1.069 \times \frac{1D}{.340}$$

$$R_c = \frac{1}{h A \eta}$$

-new- ID	$\frac{ID}{.340}$	Re	Re'^2	St	$\left(\frac{ID}{.340}\right)^2$	G	h	δ	m	l	ml	η	A	100% Rc	$\frac{1.022}{\leftarrow}$
.400	1.18	18,700	7.16	.00141	1.387	$.927 \times 10^6$	1305	.136	3.23	.440	2.06	.47	1,263	1.29	.792
.500	1.47	15,100	6.83	.00148	2.16	$.599 \times 10^6$	884	.109	2.98	.779	2.39	.41	1.573	1.79	.571
.700	2.06	10,730	6.40	.001518	4.25	$.304 \times 10^6$	478	.0776							
.450	1.325	16,700	6.97	.00145	1.75	$.738 \times 10^6$	1065	.1207	3.10	.719	2.23	.44	1.416	1.51	.677
.340	1	22,100	7.32	.00135	1	1.291×10^6	1770	.160	3.47	.542	1.88	.51	1069	1.035	.987
.600	1.764	12,520	6.61	.001528	3.11	$.415 \times 10^6$	631	.0906							



∴ Tubes must be tapered from 0.387" ID @ $x/l = 0$ to 0.363" ID @ $x/l = .25$

$$D = .387 - (.387 - .363)\frac{x/l}{.25} = .387 - .096\,x/l$$

$$.340 = .387 - .096\,x/l$$

$$x/l = \frac{.047}{.096} = .49$$

Small error in calculated value of Rc

Checking an intermediate point to see if linear taper in tube 10 is adequate.

@ $x/\ell = .10$

$$ID = .363 + \frac{3}{5}(.387 - .363) = 0.377''$$

$$\frac{R_c}{R_c} = 87\%$$

$$\therefore R_c = \frac{.001022}{.87} = .001174$$

$$\frac{R_c}{R_c + R_k} = \frac{.001174}{.001174 + .00272} = .299$$

$$q_2 = \frac{q}{2} + \frac{8}{.00272}\,\frac{.701}{1.299}(t_1 - t_2) = \frac{q}{2} + 1590\,(t_1 - t_2)$$

$$t_1 - t_2 = 1.42 - 7.1 = -\ 5.68\,°F$$

$$q_1 = 23{,}000 - 9000 = 14{,}000$$

$$q_2 = 32{,}000$$

$$t_{w_1} - t_1 = \frac{14{,}000}{3840} = 3.65\,°F$$

$$t_{w_2} - t_2 = \frac{32{,}000}{3840 \times .87} = 9.60\,°F$$

$$t_{10} - t_{1(x)} = \frac{11{,}100 + 14000}{6500} \times .10 = .386\,°F$$

$$t_{2(x)} - t_{10} = \frac{34{,}900 + 32{,}000}{6500} \times .1 = 1.03\,°F$$

$$\therefore t_{w_1} - t_{10} = 10.36\,°F$$

$$t_{w_2} - t_{10} = 10.63\,°F$$

It appears that the avg of the tube base temps is within ~0.1°F with a linear taper.

STANFORD LINEAR ACCELERATOR CENTER II (E)

Beam Pipe Cooling System

A memo Al wrote in February 1963 discussing alternative cooling tube configurations appears in Exhibit 1.

Shortly afterwards Al proposed a modified water jacket which simplified the manifolding problems considerably. This later became the final design, shown in Exhibit 2. It is described in the August 1963 Status Report (SLAC-18):

> A 12-foot-high retort furnace was put into operation during the previous quarter. The furnace was constructed to prove the feasibility of attaching the cooling tubes to the accelerator pipe by brazing and for use until the pit furnace at the site is in operation. A cooling jacket assembly was brazed successfuly to accelerator section No. 22 in the furnace. It was found that the entire assembly, including manifolds, feed tubes, fittings, and cooling tubes, could be brazed in a single operation.
>
> The water jacket on accelerator section No. 22 was modified slightly from the previous models. The inlet and outlet manifolds were located at the midpoint of the section so that the water flow from the supply manifold would flow through four feed tubes, split into eight "hair-pin" cooling tubes, and flow back through four return tubes into the return manifold. It was desirable to move the manifolds away from the accelerator section ends where accelerator support members, wave-guide transition supports, end flanges, and rf input and output wave-guides are located. The modification also cuts in half the number of manifolds and feed and return tubes required.
>
> Tests on accelerator section No. 22 in Test Stand I with up to 15 kw of average input rf power have shown that the performance of the modified water jacket arrangement is essentially equal to that of the best of the earlier water jackets. Temperatures on the accelerator fell within a 1.0°F band at the highest power level. The temperature was found to be high in the first two cavities at the input end. This effect is attributed to the heat that is dissipated in the uncooled input waveguide and then transferred into the cooler accelerator section. It is expected that cooling the input waveguide will allow us to attain a temperature uniformity on the accelerator section of $\pm$ 0.4°F at full Stage II operating conditions.

Some of the final design parameters are listed below:

Cooling tube to accelerator contact width	3/16 inch
Water flow rate	13 gpm
Water velocity	6 fps
Reynolds number	20,000
Water temperature rise (Stage I)	1.7°F
Water temperature rise (Stage II)	7.1°F
Accelerator average temperature	113°F
Cooling water inlet temperature (Stage I)	110°F
Cooling water inlet temperature (Stage II)	102°F
Water jacket pressure drop	2 psi

Al explained that the hairpin cooling tube idea had not begun to seem feasible until further experiments showed conclusively that entrance effects were negligible. "The manifold problem had been bad from the start," he said. "The space restrictions were severe. I first came up with the hairpin idea when thinking about different ways to manifold the tubes, but I was afraid the 180° bends would just double the number of places where we had a thin boundary layer and high heat transfer rates. However our tests eventually showed that with straight, counterflow tubes we would get either a drop or a rise in temperature at the ends of a pipe section, depending on power level. This is due to conduction through the waveguides. At low power the waveguides act as cooling fins and the temperatures at the ends of the section drop. At high power, heat is generated in the waveguide and in this case heat flows the other way, into the pipe. The temperature change at the ends from high to low power is two or three degrees. Later we tried water jacketing the waveguides too, but it didn't help enough to be worth the trouble. We never did measure any temperature variations we could attribute definitely to boundary layer growth."

"A side benefit of the hairpin tubes is that they're easier to handle -- only 5 feet long instead of 10. We couldn't have used more than eight hairpin tubes because the bends would've been too tight, but we would probably have decided on eight tubes anyway, for economy. From the calculations and flux plots, the .340 I.D. tube looked like a good size, not necessarily an optimum, but reasonable, and the cost of using more tubes with a higher flow rate didn't seem justified in terms of improved performance. We chose a thick walled tube for several reasons. It gives a large conduction path around the tubes and a higher fin effectiveness. We get a wide contact area, 3/16 inch, by milling the undersides of the tubes on a 2 inch radius where they're attached to the pipe. A thick wall is needed for this, and although

our studies showed the contact width isn't really critical, a large area is nice because any variations in the width or brazing alloy thickness are less significant. We can also counterbore the tubes to make connections, which allows us to keep the tube in contact with the pipe right up to the point of connection, and the heavy walled tubes are less liable to be damaged. A fully assembled 10 foot section weighs almost 500 lbs and is not too easy to handle."

"We decided to put the tubes onto the surface of the pipe without any spacers because of the lower costs and because our later tests showed this would probably work okay. The retort furnace allowed us to braze tubes on by heating the whole section up slowly, and we started getting much better results than with the soft-soldered tubes. We found both more uniform temperatures over a section and smaller variations with RF power level for the hairpin jackets as compared to the end-fed jackets, but I think most of the improvement was due to the brazing. These tests did show low pipe temperatures at the midpoint, perhaps due to entrance effects after all, but we found we could get rid of the dip in the temperature profile by raising the inlet tees between the tubes themselves and the feed pipes so they weren't in contact with the pipe surface."

By the middle of 1963 parts for all the pipe sections and jackets had been ordered. All water jacket parts were made by vendors from SLAC's stock of OFHC copper. The February 1964 Status Report (SLAC-27) gave the results of tests made to prove out the final design:

During the quarter, final design type water jackets were mounted on accelerator sections 27 through 58. The assembly technique was improved by holding the accelerator section vertical while the water jacket parts were mounted. (Previously, the assembly work had been done with the accelerator section held horizontally.)

Tests carried out during the quarter in the high power rf test stand were performed on the final water jacket design to verify its performance. The accelerator surface temperature was maintained constant during the tests. A block diagram of the temperature control arrangement is shown in Figure 6*. The water flow rate was the prescribed 13 gpm. The measured temperatures for accelerator section No. 29 are shown in Figures 7,8,9 and 10*. The temperatures were measured at average input power levels of 0, 5, 10 and 14.8 kw respectively; the last of these power levels represented the maximum practical capability of the test stand at the time these measurements were taken. All of these figures are composites of two or more test runs. Each dot on the figures represents an average value, and the vertical line shows the full range covered by the temperature values measured at that point.

* Figures 6 through 10 are contained in Exhibit 3.

The surface temperature control sensor was located on cavity No. 60. It can be seen from the figures that the temperature at that point was held constant within 0.1°F at all power levels. Other points on the accelerator section varied considerably more with power level changes. Cavity No. 60 was chosen as the control point because use of that point was found to yield the smallest phase shift with power level changes. The measured temperatures all fell in the range of 113.0°F to 114.7°F.

Al said, "We found that different sections behave the same within a few tenths of a degree."

After assembly the 10 foot sections are mounted upon supports called strongbacks, as shown in Exhibit 4. The dead soft copper of the pipe would otherwise sag excessively. The strongbacks are in turn installed on 40 foot long aluminum tubes, 2 feet in diameter, as described in the following excerpt from the February 1964 Status Report:

> After the water jacket with its manifolds is brazed onto the disk-loaded waveguide, the assembly is mounted onto the strongback. During the mounting, adjustments are made to bring the two ends and the two-thirds points of the disk-loaded waveguide straight within 0.002 inches. The strongback is designed to protect the disk-loaded waveguide and the waveguide transitions throughout the tuning and rf processing, which take place after assembly. In particular, the strongback is designed to support the disk-loaded waveguide so that it will not take on a permanent set if exposed to an acceleration of one "g" in any direction.
>
> In addition to the above functions, the strongback provides strength adequate to carry the forces imposed on the waveguide transitions by the adjoining waveguide and vacuum systems after final installation of the strongback on the 40-foot support beam. Accurately located alignment holes in the end assembly of each strongback are used for optical alignment of four strongbacks while they are being mounted on the support beam. The largest axial differential thermal expansion allowed between any of the four copper disk-loaded waveguides and the 40-foot aluminum structure is 0.320 inch. Therefore, the other functions of the strongback must not be influenced by an axial displacement of ± 0.160 inch of the disk-loaded waveguide relative to its strongback.
>
> The final design of the strongback has been completed, and specifications for this design have gone out for bids. The ten-foot beam member is an extrusion. The end assemblies will be fabricated out of castings, extrusions, and bar stock before final machining takes place. Six sets of castings for end assembly prototypes are now on order.

The beam pipe sections are aligned on the strongbacks with dial indicators. Four sections mounted on a 40 foot support beam are aligned optically to .010 inch and the entire accelerator is then aligned to 1/8 inch in 2 miles using a laser system. Provision is made for ready adjustments of up to 3 inches and up to 6 inches with additional manual adjustments, should realignment be necessary -- for example, following an earthquake.

The drawn copper cooling tubes are spaced around the surface of the pipe by four notched rings of 4 inch I.D. and 5-3/4 inch O.D., made in halves, which are machined from 6 inch O.D. copper pipe. Two of these rings are later attached to the strongback. The manifolds are machined from 2 inch bar stock, while the tees are made from 5/8 inch O.D., .465 inch I.D. tubing. Before assembly the inside surfaces of the tees are nickel plated (.0005 inch) for protection against corrosion and erosion.

During assembly of the sections, four different brazing operations are performed using copper-silver-gold brazing alloys melting at successively lower temperatures. In the first two steps, the beam pipe sections are assembled with, first, Nicoro alloy having a flow point of 1877°F, then 50-50 gold-copper flowing at 1742°F. Next, the cylinders and discs are brazed together as described in Part A. In the last step the cooling tubes and manifolds are added: first, strips of Incusil 15 (flows at 1270°F) .003 inch thick and 1/4 inch wide are spot welded to the tubes. The tubes are then staked to the half-rings and assembled to the accelerator section. The tees are added and the tubes are adjusted and wired in place. Finally the feed tubes and manifolds are positioned. Pre-formed strips and rings of the Incusil alloy are used for these parts. The section is then brazed in the retort furnace with the temperature held to 1360°F $\pm$ 10°. Other brazing methods which had been considered included salt bath dip brazing -- rejected because a very deep and expensive bath would be needed since the sections had to be held vertically -- the flame furnace and ring burner with shields so the tubes would not be overheated, and a plasma arc. Al said, "The plasma arc and the modified flame furnace never got past the talking stage, but we did try dip brazing on some short sample sections. There were other problems besides cost -- the difference in immersion times between the top and bottom and also the corrosive effects of the salt. We had to guarantee that the insides of the beam pipe cavities would have a bright, clean surface and this would have meant sealing the ends. We ended up using a hydrogen reducing atmosphere in a more nearly conventional brazing furnace."

After brazing, the water jackets are checked for leaks with a vacuum leak detector and are mounted on the strongbacks and aligned. Next the sections are tuned. A special machine was designed and built by SLAC to dimple the cavities. Four automatically controlled hydraulic rams successively squeeze each cavity while microwave measurements are made.

During this operation water is pumped through the cooling tubes with temperature and flow controlled to the design conditions. The section is then placed in an RF test stand and temperatures are checked at various power levels up to full Stage II power. Inlet and outlet water temperatures and the electrical phase shift are recorded. When the testing is completed, nitrogen gas is used to push the water out of the tubes, which are then flushed with alcohol to pick up any remaining water. Nitrogen is again used to push out the alcohol and the tubes are pumped out with a vacuum pump to vaporize any alcohol left behind. The section is then ready to be installed on the 40 foot aluminum beam. Adjacent sections are connected through stainless steel flanges with spacers for adjustment. These isolate each section. The outside of the pipe is not insulated, although it is covered by magnetic shielding. Once underway, production went at the rate of about three sections per day.

Al said, "After wrestling with the temperature control problem for a long time we finally decided that as long as we had a system that worked from a microwave standpoint, that's all we needed. When we were considering sector control I calculated that this would give us 1/2° to 1° of temperature variation. Then in view of the budget we said, why bother? -- and decided just to control the water temperature at the inlet to the whole system. No temperatures on the beam pipe section are measured at all. When we were making and testing the sections we recorded the relation between water inlet temperature and RF phase shift for all the thousand sections as a function of power level. These were averaged to get a master curve of water temperature versus RF power for zero change in phase shift and we just control the water temperature manually according to the power level."

A mixing valve is used in the inlet line to the accelerator to adjust the proportions of cold water from the cooling tower and hot water returning from the accelerator to give the desired temperature. This method gives a temperature variation in the pipe of 1.2°F from zero power to full Stage I power (3.2 kw).

Besides mixing hot water exiting from the accelerator with cold water to get the desired temperature, a system in which the return water would be flash evaporated in a tank and re-condensed to come out at the desired temperature was also considered. Several 20 to 30 kw capacity systems of each type were built and tested. Performance was found to be about equal in terms of both steady state operation and transient response to a change in power level. The mixing valve system was chosen because it could be built largely with standard off-the-shelf components while most of the parts for the flash tank system would have had to be specially manufactured.

Al said, "During RF tests we found that we could determine the average bulk metal temperature of the whole pipe with greater accuracy (± .01°F) by measuring the electrical phase shift than we were ever able to do by direct measurements with thermocouples (± 0.1°F)."

DATE: February 5, 1963

TO : A. L. Eldredge

FROM : A. V. Lisin

RECEIVED

FEB 6 1963

ACCELERATOR STRUCTURES DEPT.

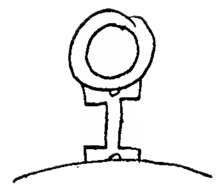
SUBJECT: Summary of Several Longitudinal Tube Type Water Jackets

On the attached sheets, I have briefly described each of the longitudinal tube type of cooling jackets which look feasible and which do not cost much over $500. The ability of each type of jacket to maintain a uniform accelerator temperature and the effects of tolerances on the temperature uniformity are described. All temperature non-uniformities reported are for Stage II full power operation, and assume a cooling water flow rate of 13 gpm. These non-uniformities are directly proportional to power level.

Cost estimates are based on the best available information (cooling tube cost is based on Anaconda's price. Copper bar cost is extrapolated from other Anaconda prices. Cleaning times were obtained from J. Pope. Inspection times were arrived at jointly with E. Jones. Assembly times were arrived at jointly with H. Söderstrom. Machining costs are based on engineering estimates obtained from outside shops. The SLAC hourly labor cost was assumed to be $7.00).

I suggest that the final choice of water jacket design be made from one of two designs. If temperature is all important and cost is secondary, the constant diameter tube on the 'I' spacer offers reliability and temperature uniformity at a cost of $377.00 per water jacket assembly. If cost is more important than temperature uniformity, the constant diameter cooling tubes brazed directly to the accelerator offer a highly reliable system at about $219.00 per water jacket.

Constant Diameter Tube with 'I' Spacer



Description

Cooling tube is 1/2" OD x .080" wall. Spacer is machined from 5/8 x 3/8 bar. Edges are machined to conform to accelerator pipe and cooling tube curvatures. Brazing alloy wire grooves are machined in same operations. 1/4" high web is machined to varying thickness to correct for water temperature rise and entry effects, yielding a constant accelerator wall temperature. Web thickness varies as shown in SK 751-025-R0.* The heat transfer registance added by the web would raise the accelerator temperature by 2° to 115°F. Tolerance buildup can result in a ± .80°F temperature variation in the accelerator. Total cost of complete water jacket assembly is $377. The chances of this water jacket performing as predicted are very good.

Tolerance Effects

Cooling tube ID	.340 ± .005	± .07°F
Water flow rate	13 gpm ± 2%	± .13°F
Tube wall thickness	.080 ± .0035	± .08°F
Web height	.250 ± .003	± .05°F
Web thickness	.06 (min) ± .003	± .15°F
Braze joints widths	.34 ± .031	± .30°F
Braze joints thickness	.001 ± .001	± .02°F
Maximum possible temperature variation		± .80°F

Costs

Materials:

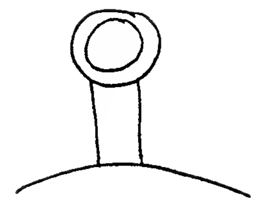
Bar (3/8 x 5/8) .90 lb/ft x 80 ft x $1.20/lb	87
Tubes (1/2 OD x .080 wall) .405 lb/ft x 80 ft x $1.82/lb =	59
Alloy (.035 dia. Incusil) .062 oz/ft x 160 ft x $2.10/oz =	21
Furnace (power and gases)	40
Manifolds and fittings	30
Temperature Sensor	20
Total material cost	$257

* Page 12 of this exhibit.

Labor:		
Machining spacer	$7.50 ea x 8	60
Inspection	.05 hr/tube x 8	3
	.05 hr/spacer x 8	3
Cleaning	.01 hr/part x 16	1
Assembly	2 man hrs	14
Brazing	1 man hr	7
Hydrostatic test and adjust	3 man hrs	21
Calibrate and wire temp. sensor	.5 man hrs	4
Dry and seal tubes	1 man hr	7
Total labor cost		$120
Total cost		$377

Constant Diameter Tube with Tapered Rectangular Spacer

Description



The cooling tube is 1/2" OD x .080 wall. The spacer is machined from 1/4 x 7/8 bar. The spacer height varies along its length in a manner which compensates for the water temperature rise and entrance effects. The machined edges are made concave to fit the cooling tube curvature and accelerator curvature. The dimensions of the tapered spacer are shown in Figures 1 and 2.* Brazing alloy in sheet form would be used. The heat transfer resistance added by the spacer would cause the accelerator operating temperature to increase by 2°F to 115°F. Tolerance build up can result in a ± .81°F temperature variation in the accelerator. The total cost of such a water jacket assembly is $377. The chances of this water jacket performing as predicted are quite good.

Tolerance Effects

Cooling tube ID	.340 ± .005	± .07°F
Water flow rate	13 gpm ± 2%	± .13°F
Tube wall thickness	.080 ± .0035	± .08°F
Spacer height	1.0 ± .002	± .02°F
Spacer width	.25 + .000 - .005	± .09°F
Braze joints widths	.250 ± .031	± .40°F
Braze joints heights	.001 ± .001	± .02°F
Maximum possible temperature variation		± .81°F

Costs

Materials:

Bar (1/4 x 1-1/8) 1.09 lb/ft x 80 ft x $120/lb =	104
Tubes (1/2 OD x .080 wall) .405 lb/ft x 80 ft x $1.82/lb -	59
Alloy (.005 x 5/16 sheet) .00792 oz/sq. in. x .312 x 160 x 12 sq. in. x $3.50/oz =	17
Furnace (power and gases)	40
Manifolds and fittings	30
Temperature sensor	20
Total material cost	$270

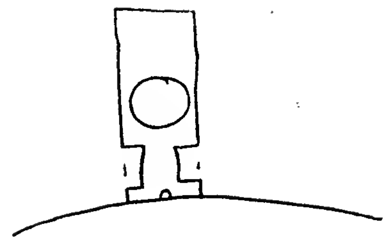
* Not included. These figures are the same as Figures 4 and 5 on pages 9 and 10 of Exhibit 5 in Part D (Al's previous memo on water jackets).

Labor:

Machining spacer	$4.50 x 8	36
Inspection	.05 hr/tube x 8	3
	.05 hr/spacer x 8	3
Cleaning	.01 hr/part x 16	1
Assembly	3.5 man hrs	25
Brazing	1 man hr	7
Hydrostatic test and adjust	3 man hrs	21
Calibrate and wire temp. sensor	.5 man hrs	4
Dry and seal tubes	1 man hr	7
Total labor cost		$107
Total cost		$377

Integral Tube and Spacer

Description



Coolant passage and variable spacer are made from a single piece. The extruded piece is 1 1/4" x 1/2" with 0.340 hole on the center. One edge of the piece is machined to conform to curvature of accelerator pipe wall and to accept brazing alloy wire. The 1/4" high web is machined to varying thickness to correct for water temperature rise and entry effects, yielding a constant accelerator wall temperature. Web thickness varies in same way as for seperate spacer (part no. SK 751-025-RO). Because of the heat transfer resistance of the web the accelerator temperature will have to be increased by $2^{\circ}F$ to $115^{\circ}F$. Tolerance buildup can result in a $\pm$ $.59^{\circ}F$ temperature variation in the accelerator wall. Total cost of complete water jacket assembly is $504. The chances of the water jacket performing as predicted are very good.

Tolerance Effects

Flow passage ID	.340 $\pm$ .005	$\pm$ $.07^{\circ}F$
Water flow rate	13 gpm $\pm$ 2%	$\pm$ $.13^{\circ}F$
Wall thickness	.080 $\pm$ .005	$\pm$ $.03^{\circ}F$
Web height	.250 $\pm$ .003	$\pm$ $.05^{\circ}F$
Web thickness	.06 (min) $\pm$ .003	$\pm$ $.15^{\circ}F$
Braze joint width	.40 $\pm$ .031	$\pm$ $.15^{\circ}F$
Braze joint height	.001 $\pm$ .001	$\pm$ $.01^{\circ}F$
Maximum possible temperature variation		$\pm$ $.59^{\circ}F$

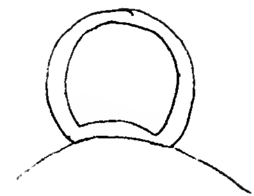
Costs

Materials:	
Bar (1 1/4 x 1/2 OD x .340 ID)	
2.08 lb/ft x 30 ft x $1.80/lb =	300
Alloy (.035 dia. Incusil)	
.062 oz/ft x 80 ft x $2.10/oz =	10
Furnace (power and gases)	40
Manifolds and fittings	30
Temperature sensor	20
Total material cost	$400

Labor:		
Machining	$6.00 ea x 8	48
Inspection	.05 hr/tube x 8 tubes	3
Cleaning	.01 hr/part x 8	1
Assembly	1.5 man hr	10
Brazing	1 man hr	7
Hydrostatic test and adjust	3 man hrs	21
Calibrate temp. sensor and wiring	.5 man hr	4
Dry and seal tubes	1 man hr	7
Total labor cost		$101
Total cost		$501

Tapered Tube

Description



1/2" OD x .080 wall cooling tubes are expanded by hydroforming (in two steps with an intermediate anneal) in a tapered die. The tube diameter and wall thickness would vary in a manner which will just correct for the cooling water temperature rise and entrance effects. The tube diameter variation is shown in Figure 3. A concave surface to conform to the accelerator wall curvature would be formed into the tube in the tapering process. Sheet alloy would be used for brazing. Tolerance buildup can result in a ± .95°F temperature variation in the accelerator. The total cost of such a water jacket assembly is $358. This design would be less reliable than the previous two because of the uncertainty in flow behavior in a constantly converging flow passage and because of the low Reynolds' number in the entrance region.

Tolerance Effects

Flow passage ID	~7 (max) ± .005	± .20°F
Water flow rate	13 gpm ± 2%	± .13°F
Wall thickness	.040 (min) ± .010	± .25°F
Braze joint width	.187 ± .047	± .35°F
Braze joint height	.001 ± .001	± .02°F
Maximum expected temperature variation		± .95°F

Costs

Materials:	
Tubes (1/2 OD x .080 wall)	
.405 lb/ft x 80 ft x $1.82/lb =	59
Alloy (.0015 x 1/4 sheet)	
.00792 oz/sq. in. x .25 x 80 x 12 sq. in. x $3.50/oz =	7
Furnace (power and gases)	40
Manifolds and fittings	30
Temperature sensor	20
Total material cost	$ 156

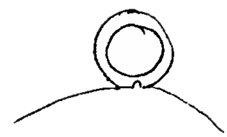
Labor:

Tapering tubes	$16.00 ea x 8	128 *
Inspection	.1 hr/tube x 8	6
Cleaning	.01 hr/tube x 8	1
Assembly	3 man hrs	21
Brazing	1 man hr	7
Hydrostatic test and adjust	4 man hrs	28
Calibrate and wire temp. sensor	.5 man hr	4
Dry and seal tubes	1 man hr	7
Total labor cost		$202
Total cost		$358

* Assuming forming can be done in two steps. A third step would increase cost by $64.

Constant Diameter Tube

Description



The cooling tube is 1/2" OD x .080 wall everywhere. A concave surface is machined into the tube wall to conform to the accelerator pipe curvature. A .030 deep groove for the brazing alloy wire is milled in the same operation. No correction is made for the water temperature rise or entrance effects. Over most of the accelerator length, the temperature is within 0.9°F. Over the 3 or 4 end cavities the temperature drops so that it is 1.7°F low at the accelerator ends. The tolerance buildup can result in an additional temperaturevariation in the accelerator of $\pm .59^{\circ}$F. The cost of such a water jacket is $219. The chances of this type of jacket of performing reliably and repeatably are very good.

Tolerance Effects

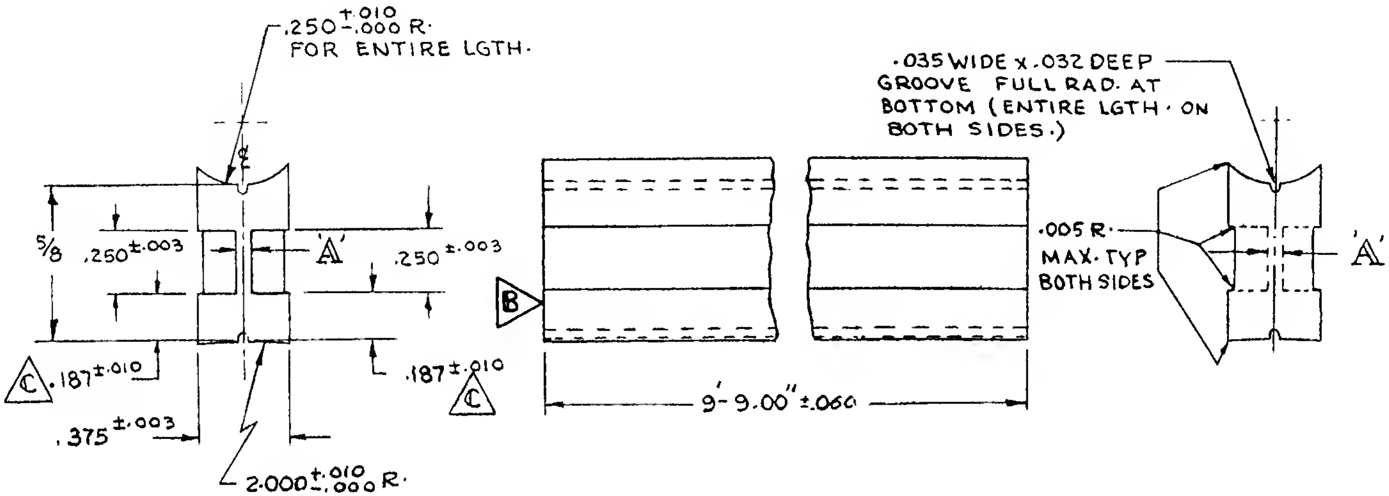
Cooling tube ID	$.340 \pm .005$	$\pm .07^{\circ}$F
Water flow rate	13 gmp $\pm$ 2%	$\pm .13^{\circ}$F
Tube wall thickness	$.080 \pm .0035$	$\pm .08^{\circ}$F
Braze joint width	$.160 \pm .031$	$\pm .30^{\circ}$F
Braze joint height	$.001 \pm .001$	$\pm .01^{\circ}$F
Maximum expected temperature variation		$\pm .59^{\circ}$F

Costs

Materials:

Cooling tubes (1/2 OD x .080 wall)	
.405 lb/ft x 80 ft x $1.82/lb =	59
Alloy (.030 dia. Incusil)	
.406 oz/ft x 80 ft x $2.10/oz	8
Furnace (power and gases)	40
Manifolds and fittings	30
Temperature sensor	20
Total material cost	$157

Labor:		
Machining	.02 hr/tube x 8	1
Inspection	.05 hr/tube x 8	3
Cleaning	.01 hr/tube x 8	1
Assembly	2.5 man hrs	18
Brazing	1 man hr	7
Hydrostatic test and adjust	3 man hrs	21
Calibrate temp. sensor	.5 man hrs	4
Dry and seal	1 man hr	7
Total labor cost		$62
Total cost		$219



DIST. FROM THIN END 'B' IN INCHES ±.030	DIM 'A' IN INCHES ±.003
0.00	.054
2.34	.062
4.68	.068
7.02	.073
9.36	.076
11.70	.078
23.40	.085
35.10	.093
46.80	.104
58.50	.117
70.20	.134
81.90	.157
93.60	.189
105.30	.238
117.00	.320

1. DIM. 'A' VARIES SMOOTHLY ALONG LENGTH OF PART AS SHOWN IN TABULATION. TOL. ON 'A' = ±.003.
2. THE WEB (DIM. 'A' IS TO BE CENTERED ON VERTICAL ℄ WITHIN ±.030.
3. DIMS. C TO BE EQUAL WITHIN ±.003.

FEB 5 1963

		O.F.H.C. COPPER (SUPPLIED BY S.L.A.C.)
	QUA.	MAT'L

STANFORD LINEAR ACCELERATOR—M
U.S. ATOMIC ENERGY COMMISSION

ENGR. [signature] CHK'D.
DFTS. R. Laurie APPV'D.

TITLE SPACER—CONST. GRAD. WATER JACKET

DATE 1-30-63
SCALE 2:1

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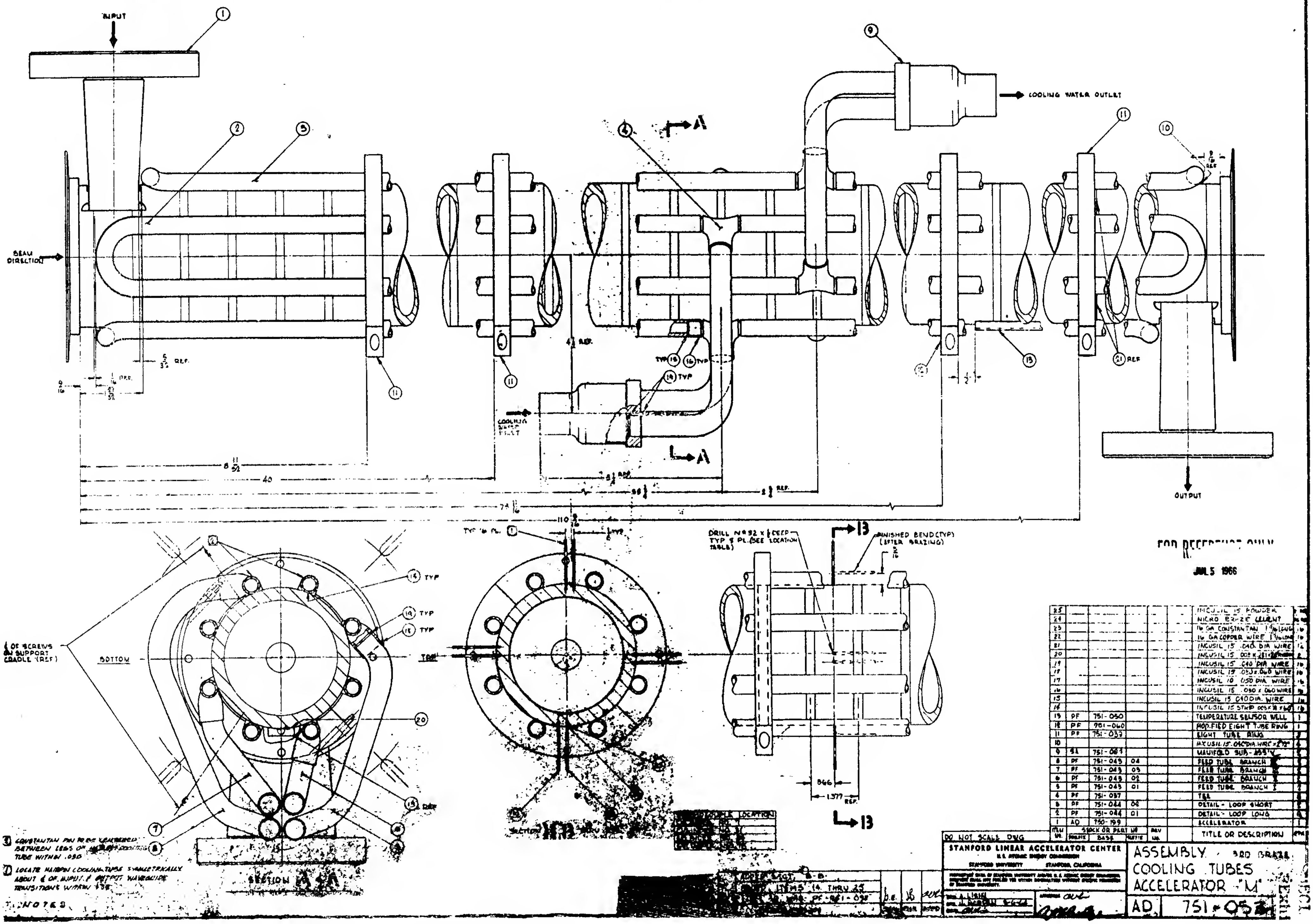

Exhibit 2: Hairpin Cooling Tube Assembly Drawing.

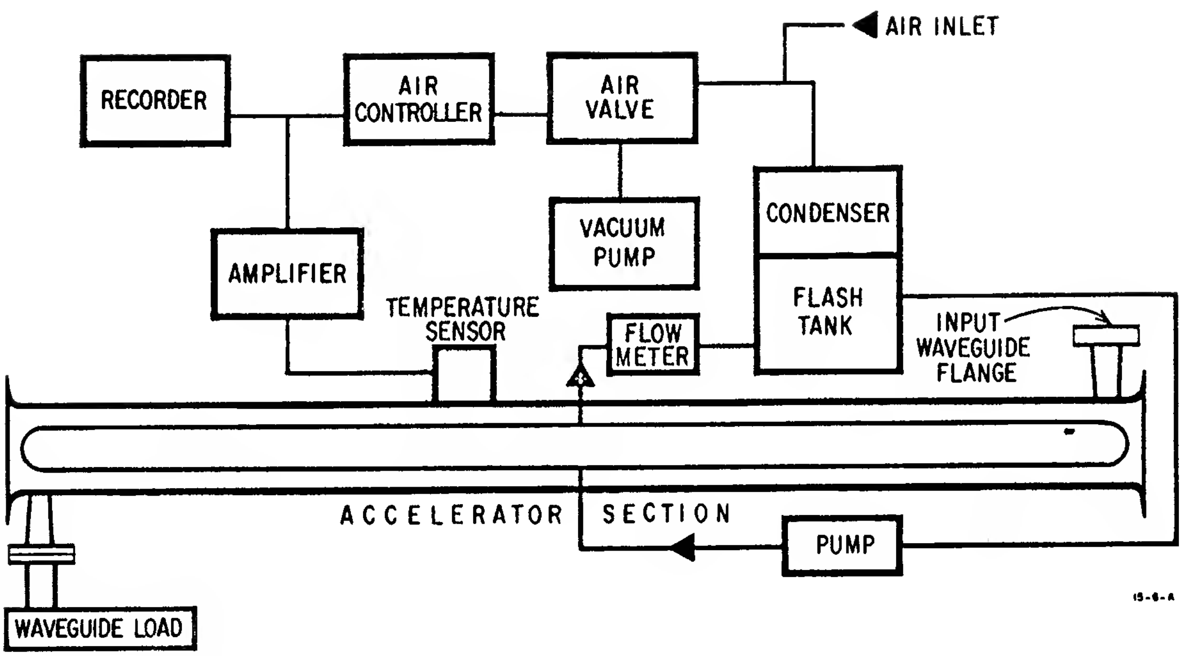

FIG. 6--Temperature control system for accelerator section.

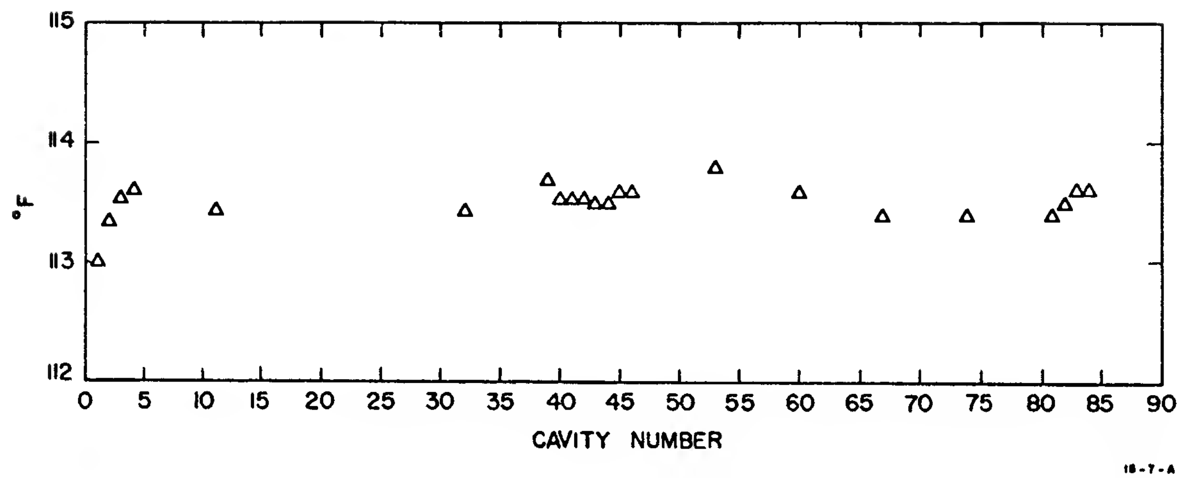

FIG. 7--Measured temperatures on accelerator section - 0 average input rf power.

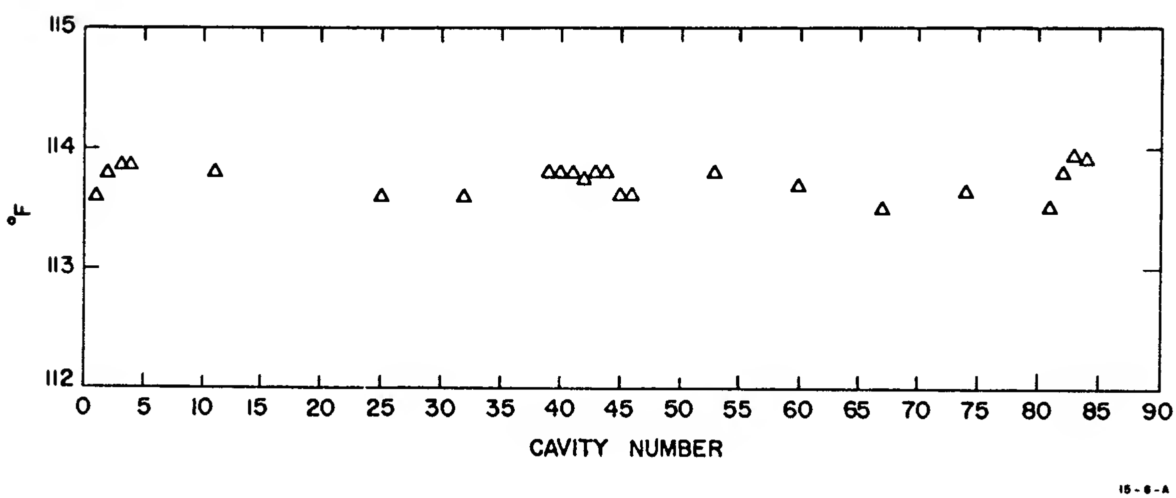

FIG. 8--Measured temperatures on accelerator section - 5 kW average input rf power.

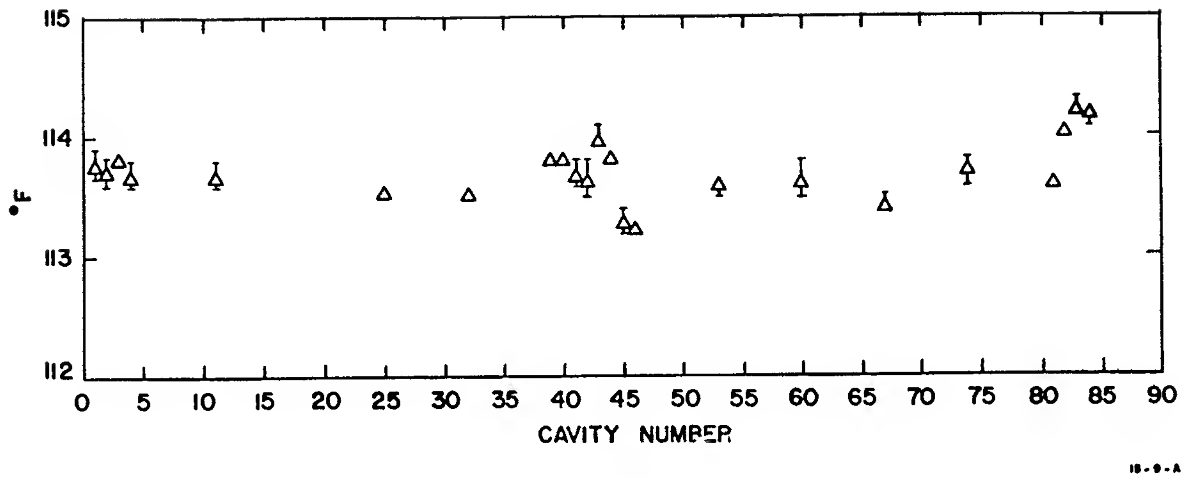

FIG. 9--Measured temperatures on accelerator section - 10 kW average input rf power.

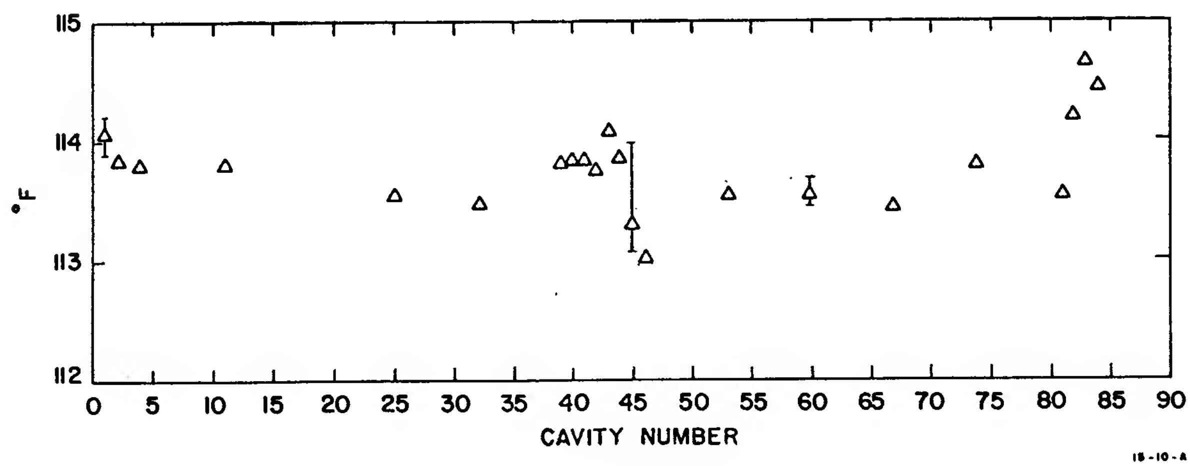

FIG. 10--Measured temperatures on accelerator section - 15 kW average input rf power.

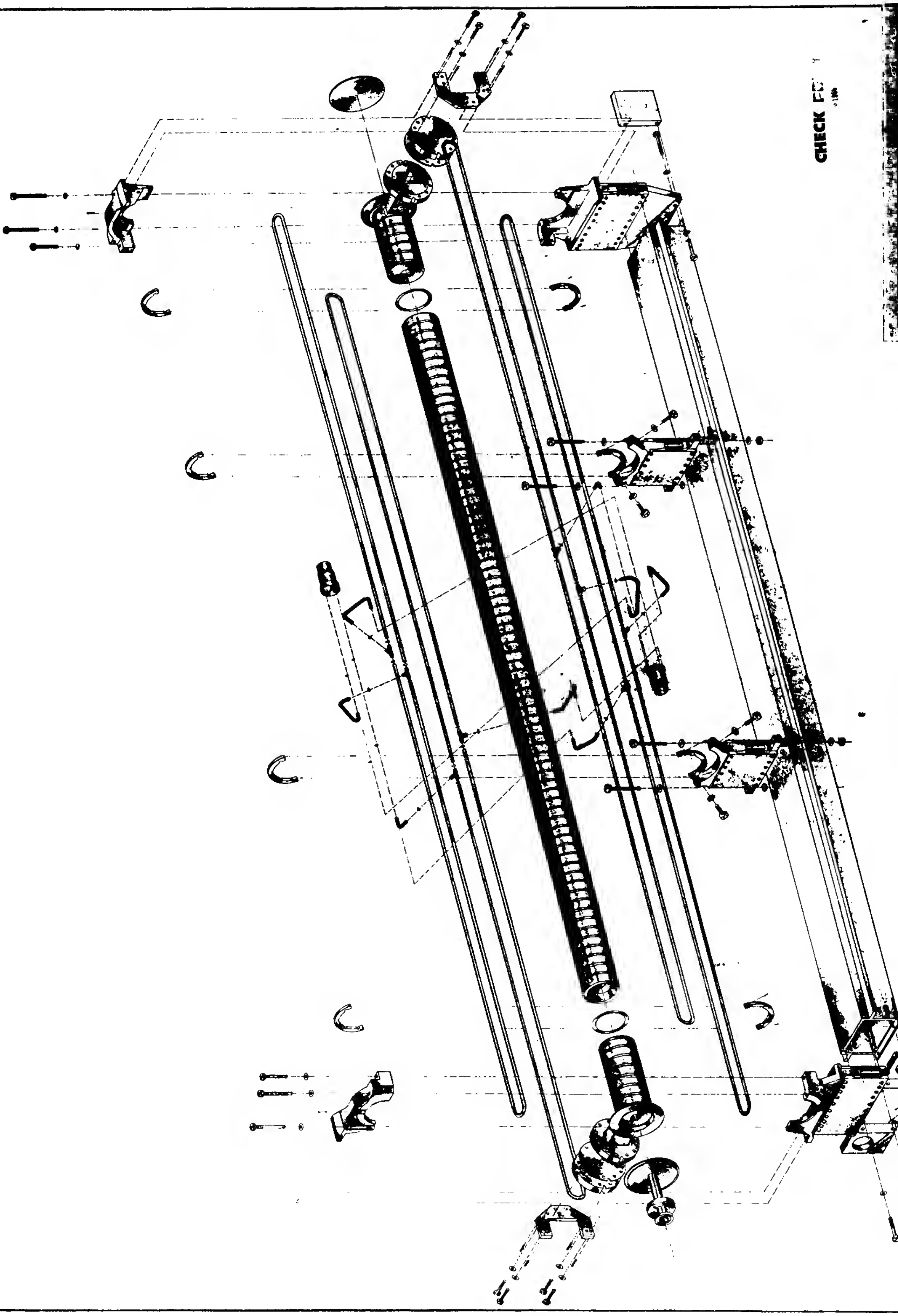

Exhibit 4: Accelerator Section and Strongback.